Public Works Canada
Solar Programs Office

SOLAR DESIGN COURSE MANUAL

Part I: FUNDAMENTALS

May 1982

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This document is not a departmental publication. Do not cite as reference or catalogue in library.

Prepared for

PUBLIC WORKS CANADA

SOLAR PROGRAMS OFFICE

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CONTRACT NO. 330 - 300 - 7

SOLAR DESIGN COURSE MANUAL

PART 1 - FUNDAMENTALS

FUNDAMENTALS
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CHAPTER 1

INTRODUCTION

1.1 COURSE CONTENT

This course in Solar Energy System Design has as its objective the training of engineers and others who will be involved in Public Works Canada's PUSH program. It is directed towards both the private and the public sectors and towards staff who will be supervising and administering PUSH projects as well as towards design engineers.

The course will be presented in two parts. Part I, "Fundamentals", will cover solar radiation, collector heat transfer, an introduction to the design of liquid and air systems, collector durability, and system components. Part II, on "Advanced Topics", will cover design methodology, site selection, simulation techniques, design approach, and inspection and operational check-out. A major element of the course on "Advanced Topics will be a practice session on the use of F-CHART 4.0 through a computer terminal.

The course is offered in two independent segments, for the benefit of those who may already be familiar with solar energy fundamentals, or who do not have time to take the whole course at once.

The design of space heating systems and of domestic and process water heating systems will be dealt with, but the emphasis will be on the water heating systems since they are the more common, as well as the more economic, applications of solar energy in Canada.

1.2 CLASSIFICATION OF COLLECTORS AND SYSTEMS

Solar collectors may be either air cooled or liquid cooled. This provides the major distinction not only between collectors, but between system types. Air cooled collectors are normally glazed flat plates. Liquid cooled flat plate collectors may be glazed or unglazed. Another type of liquid cooled collector is the evacuated tube, in which convective heat loss is prevented by an evacuated annulus around the absorber tube. There are many other types of liquid cooled collectors with various concentration arrangements, but no concentrating collector is approved under the PUSH program.

Apart from collector type, the major distinctions between solar energy systems are between active and passive systems, between applications of the solar energy collected, and between methods of freeze protection. Only active systems are elegible for installation under the PUSH Program, and only active systems will be dealt with in this course. The major applications of solar energy in Canada are:

1. swimming pool heating
2. domestic and commercial service water heating
3. industrial process water heating
4. make-up air preheating
5. space heating
6. drying of agricultural crops or other materials.

Solar cooling is also possible in Canada, although it is unlikely to be economic.

Of the above applications, drying and air preheating are best handled by air systems, space heating is usually best handled by air systems, and the other three applications are best handled by liquid systems.

Air systems need no freeze protection, of course. Some liquid systems, particularly swimming pool heaters, use the simplest of all freeze protection systems, shut-down and drainage over the winter. For those systems which must operate in freezing weather, freeze protection can be provided by using a liquid which does not freeze at ambient conditions (such as a glycol solution or an organic liquid) to cool the collectors, or by automatic drainage of the collectors when freezing is a possibility. If the latter technique is used the collectors may "drain back" into a storage tank or "drain down" out of the system. In some climates, recirculation of warm water from storage is used to prevent freezing, but this is not feasible in Canada's cold winters.

1.3 HISTORICAL REVIEW

The concept of solar heating has been well known since the beginning of the century and, before natural gas became available in the late 1940's, solar domestic water heaters were common in Florida and Southern California. They have been used for many years in Australia and Israel. Early research and development work on solar space heating was done at MIT in the 1940's and 1950's, but it was not until the sharp rise in oil prices in the early 1970's that there was renewed interest and activity in solar energy in North America.

The first active solar space heating system in Canada was built by Erich Hoffman in Surrey, B.C. His system consisted of 37 m^2 of collector with 3600 L of heat storage in tanks in a small room in the basement. To heat this house, a door is opened to allow the circulation of warmed air.

About 1975 the Canadian Government began supporting solar heating in Canada. Greg Allan's Gananoque house received a $15,000 grant. It has 22 m^2 of collectors with heat storage in 1800 kg of wax in plastic bags. Another Government supported project was the Meadowvale house at Mississauga, Ontario. This house, with 125 m^2 of floor area, has 64 m^2 of liquid cooled collectors and heat storage in a 22730 L concrete water tank.

Not all solar energy projects of this period were Government supported. Martin Stanley-Jones built a highly energy conserving 172 m^2 house near Ottawa, and heated it with 19 m^2 of Sunworks collector, storing heat in 22700 L of water. Rick Symmes built a plug-on air collector with pebble bed storage, which cost him $1,300 in scavenged material and which he estimated would meet 25% of his heating load.

Two ambitious Government funded projects of the late 1970's were the new Alchemy Ark near Charlottetown and Provident House. In the former, 79 m^2 of water cooled collector provided heat to 95,500 L of storage, for space heating. In the latter, the concept of annual storage was applied for the first time in Canada. Provident House has 260 m^2 of floor area and is heated by 67 m^2 of water cooled collector panel. Heat storage is in a 273,000 L tank, filling the basement, which is designed to store heat from the summer to provide 100% of the winter heating needs.

The Canadian Government concluded, in 1978, that the approach they had taken to supporting solar energy was not leading to the development of a Canadian Industry. Therefore they set up the Solar Programs Office within the Public Works Canada, and funded the PASEM (Program of Assistance to Solar Equipment Manufacturers) and PUSH (Purchase and Use of Solar Heating) programs. PASEM, with a budget of $4.1 million, funded the development of 21 new solar products by ten Canadian solar equipment manufacturers. PUSH with a budget of $125 million, aims to use this equipment to provide space and water heating to Federal Government buildings. About 500 projects had been funded by 1981, ranging from domestic hot water systems up to systems with over 500 m^2 of collector costing over a quarter million dollars.

Another major solar energy program currently underway in Canada is the Energy, Mines and Resources Domestic Hot Water (DHW) Demonstration Program, which is installing about 900 residential solar DHW systems across Canada.

Present activity in the Canadian solar industry is concentrated in these Federal Government programs, and in several program funded by provincial governments. At present, private installations of solar energy systems are mostly of swimming pool heaters, which offer the purchaser a reasonably short payback period at today's escalating energy prices.

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CHAPTER 2

SOLAR RADIATION

2.1 INTRODUCTION

The aspects of solar radiation covered in this chapter are:

1. the physical nature of solar radiation,
2. the geometry of solar radiation,
3. the availability of solar radiation data, and
4. the conversion of data to the required form.

A more thorough discussion of this material is covered in Chapters 1, 2 and 3 of "Solar Engineering of Thermal Processes" by Duffie and Beckman (D and B). For clarity, the nomenclature used here is that used by D and B.

2.2 THE NATURE OF SOLAR RADIATION

Thermal radiation is electromagnetic, like x-rays, radio, gamma radiation, etc., and follows the wave equation

$$C = C_0/n = \lambda\nu \qquad \text{(Eqn 2.2.1)}$$

The basic radiation properties of a material object are:

1. absorptance $\alpha = q(\text{absorbed})/q(\text{incident})$ (Eqn 2.2.2)
2. reflectance $\rho = q(\text{reflected})/q(\text{incident})$ (Eqn 2.2.3)
3. transmittance $\tau = q(\text{transmitted})/q(\text{incident})$ (Eqn 2.2.4)

The words transmittance, absorbtance and reflectance are used rather than transmissivity, absorbtivity and reflectivity because these are properties of specific objects rather than of particular materials. A pane of glass of a particular thickness will have a transmittance depending on its thickness, while the transmissivity is a property of the material of which the pane is made and is independent of thickness.

It might seem that the words reflectance and reflectivity should be interchangable since the phenomenon they measure is independent of the geometry of the body behind the surface. However, surface roughness, oxide coatings, etc., change the reflectance of surfaces, so the word reflectivity is reserved for idealised smooth, clean surfaces, so that it depends only on the surface material, while reflectance is applied to real surfaces.

A surface is called black if it can absorb all incident radiation for all wavelengths and for all angles of incidence; that is, it has an absorbtance of 1. A black body at the absolute temperature T will emit an amount of energy per unit area equal to

$$E_b = \sigma T^4 \qquad \text{(Eqn 2.2.5)}$$

where is the Stefan Boltzman constant.

$$\sigma = .1714 \times 10^{-8} \text{ BTU/ft}^2 \text{ hr } {}^0R^4$$

$$= 5.6697 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$$

No surface can emit more energy per unit area than a black body at the same temperature. Emittance is defined as

$$\varepsilon(T) = E(T) \ / \ E_b(T) \qquad \text{(Eqn 2.2.6)}$$

Note that emittance is a function of temperature, as are the other radiation properties.

Kirchoff's law states that $\varepsilon_\lambda(T) = \alpha_\lambda(T)$ for any surface. However, this does not imply that $\varepsilon = \alpha$ over the whole range of wavelengths for any particular temperature. Thus it is possible to produce a selective surface, which has an absorbtance of 0.9 for solar radiation, concentrated in the wavelength range from 0.2 to 1.0 microns, and has an emittance of 0.1 when it is at 300 K and is emitting radiation in the range from 7 to 15 microns. The wavelength distribution of blackbody radiation is Planck's law:

$$E_{b\lambda} = \frac{2\pi h C_o^2}{\lambda^5 (e^{hC_o/\lambda kT} - 1)} \qquad \text{(Eqn 2.2.7)}$$

where h is Planck's constant

k is Boltzman's constant

$C_1 = 2\pi h C_o^2 = 3.7405 \times 10^{-16}$ W m²

$C_2 = hC_o/k = .0143879$ m K

This distribution is plotted on a log scale for three temperatures in Figure 2.2.1.

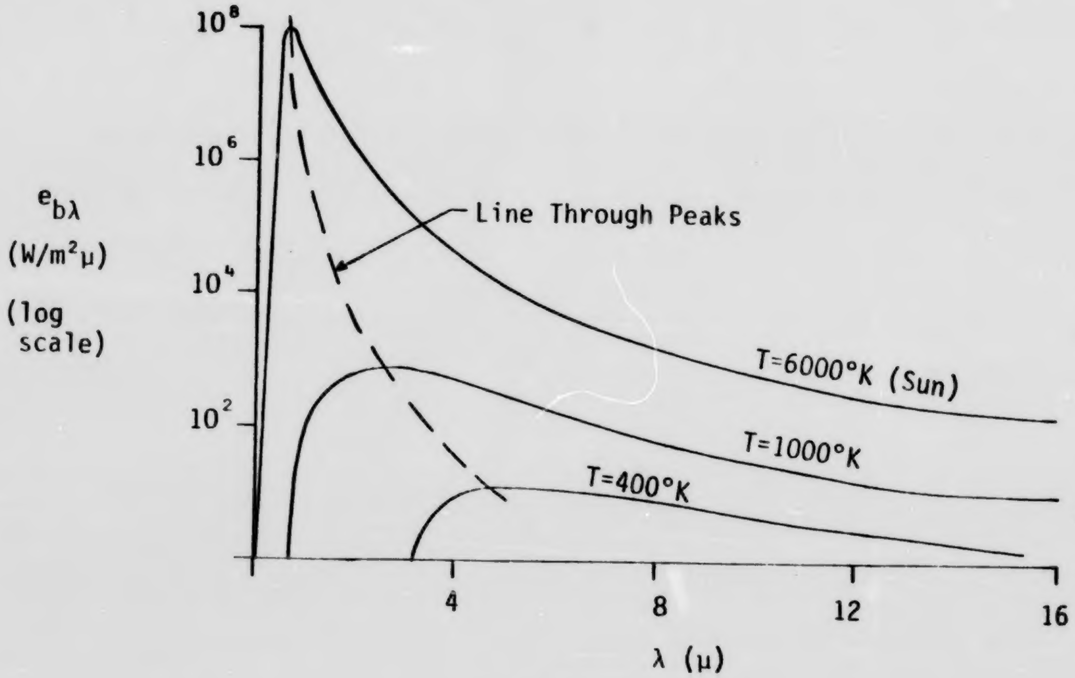


FIGURE 2.2.1

DISTRIBUTION OF EMISSIVE POWER WITH WAVELENGTH

Wien's displacement law

$$\lambda_{max} T = 2897.8 \mu K \qquad \text{(Eqn 2.2.8)}$$

describes the line throughout the peaks.

A normalized plot of these three curves is shown in Figure 2.2.2.

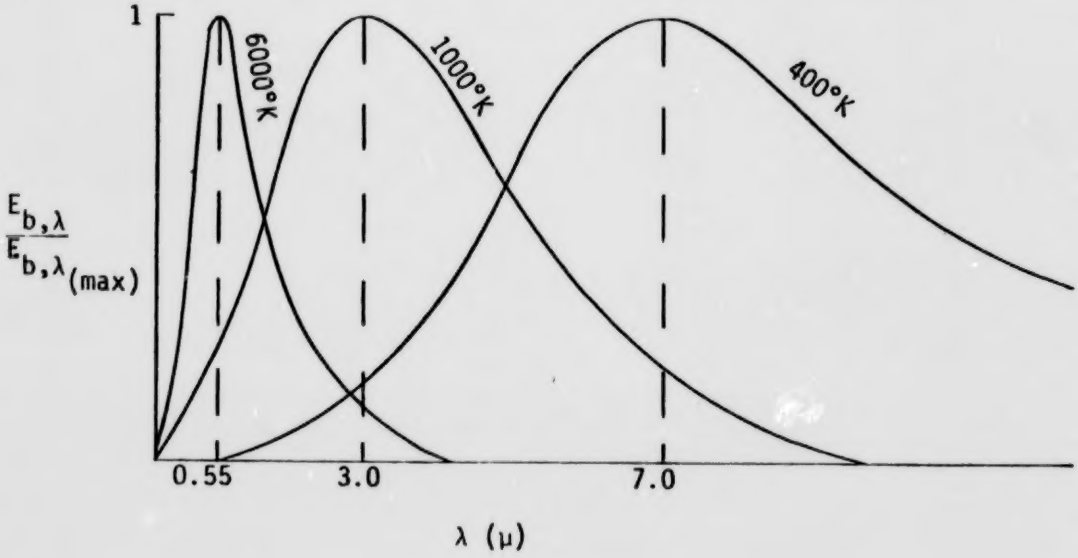


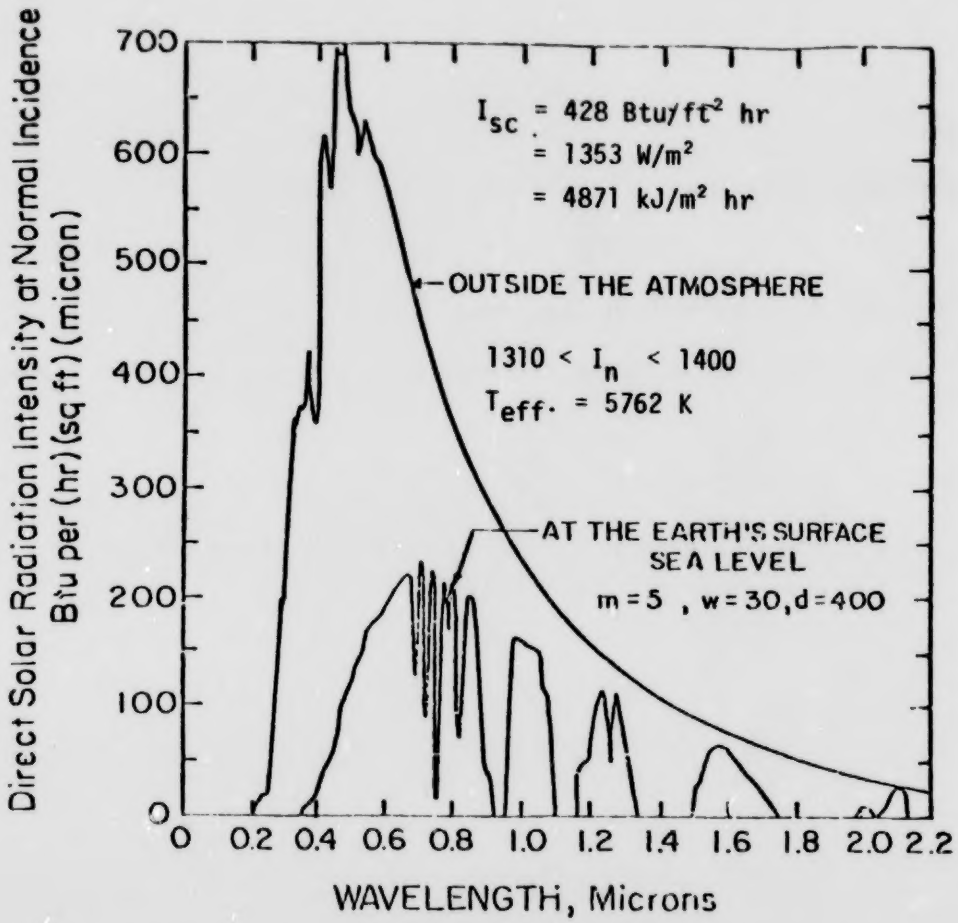
FIGURE 2.2.2

NORMALIZED DISTRIBUTION OF EMISSIVE POWER
AS A FUNCTION OF WAVELENGTH

Integration of the equation for $E_{b\lambda}$ yields the areas under these curves

$$E_b = {}_0\int^{\infty} E_{b\lambda} d\lambda = \sigma T^4 \qquad \text{(Eqn 2.2.9)}$$

No real surface emits as a black body. The sun's emission approximates that of a black body at 5762 K, but, as Figure 2.2.3 shows, it is not completely smooth. By the time the radiation penetrates to the earth's surface its frequency distribution has been greatly influenced by absorbtion by ozone, water vapour and carbon dioxide and by dust particles.



SPECTRAL DISTRIBUTION OF DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR THE UPPER LIMIT OF THE ATMOSPHERE AND AT THE EARTH'S SURFACE FOR CONDITIONS AT SEA LEVEL, WHERE :

$$m \equiv \text{Air mass} = \frac{\text{length of path of sun rays through the atmosphere}}{\text{shortest path of sun rays through the atmosphere}}$$

$w \equiv$ Depth of atmospheric precipital water — mm

$d \equiv$ Dust particles per cc

$m = 5 \equiv$ The suns altitude of 11° and Transmissivity ÷ 0.28.

RADIATION SPECTRUM

FIGURE 2.2.3

If there were no atmosphere the solar intensity at the earth's surface would be 1353 w/m^2 with a variation of ±3% depending on the time of year. This is because the earth's distance from the sun varies through the year, so that the solar intensity must also vary according to the inverse square law. Where no measured data is available, it is possible to estimate the clear sky radiation from the solar constant, using the method outlined in Section 2.8 of D and B.

2.3 THE GEOMETRY OF SOLAR RADIATION

Knowing the strength of the sun's beam at the earth's surface does not provide sufficient information to calculate the energy falling on a collector. This is because

1. the collector is not normal to the beam, and
2. much of the radiation falling on the collector is diffuse radiation from the sky and the ground.

The geometric problem is the simpler of the two, depending only on mathematical analysis. Figure 2.3.1 shows the nomenclature for this analysis.

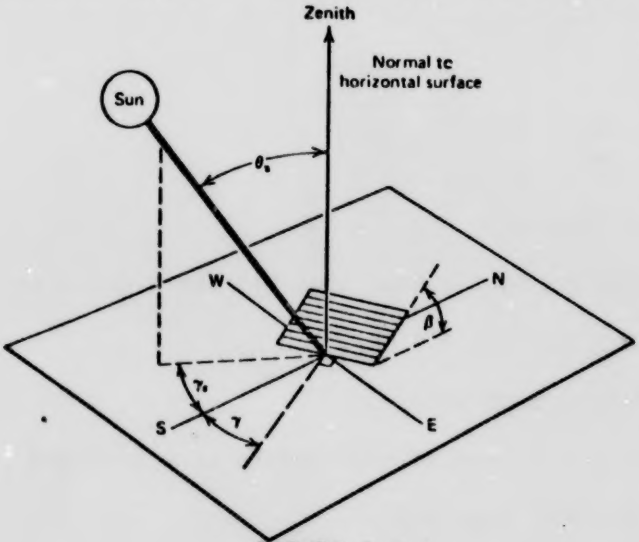


FIGURE 2.3.1

ZENITH ANGLE, SLOPE, SURFACE AZIMUTH ANGLE AND SOLAR AZIMUTH ANGLE FOR A TILTED SURFACE

The equation for the angle of incidence, Θ, of the solar beam on a surface is:

$$\begin{aligned}\text{Cos}\,\Theta = {} & \sin\delta \sin\phi \,\text{Cos}\,\beta \\ & - \sin\delta \,\text{Cos}\,\phi \,\text{Sin}\,\beta \,\text{Cos}\,\gamma \\ & + \cos\delta \cos\phi \,\text{Cos}\,\beta \,\text{Cos}\,\omega \\ & + \cos\delta \sin\phi \sin\beta \,\text{Cos}\,\gamma \,\text{Cos}\,\omega \\ & + \cos\delta \sin\beta \sin\gamma \,\text{Sin}\,\omega \end{aligned} \qquad \text{(Eqn 2.3.1)}$$

where ϕ = latitude (N = +)

δ = declination (N = +)

β = slope of surface (horizontal = 0)

γ = surface azimuth (S=0, W = +90°)

ω = hour angle (11:00 = -15°
13:00 = +15°)

The declination is given by

$$\delta = 23.45° \,(\text{Sin } 360)\left(\frac{284 + n}{365}\right) \qquad \text{(Eqn 2.3.2)}$$

where n = day of the year.

The time for determining the hour angle is sun time, given by

$$\tau = \text{standard time} + E + 4\left(L_{ST} - L_{LOC}\right)$$

where E = the equation of time

L_{ST} = the longitude on which standard time is based

L_{LOC} = the local longitude

The equation of time is shown in Figure 2.3.2, and is given by

$$E = 9.87 \sin(2B) - 7.53 \cos(B) - 1.5 \sin(B) \qquad \text{(Eqn 2.3.3)}$$

where $B = 360\,(n-81)/364$

and n = day of the year.

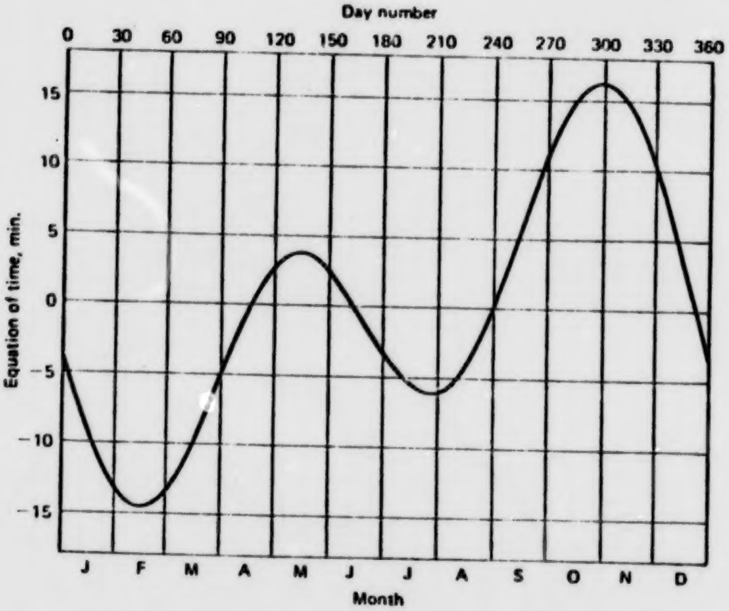
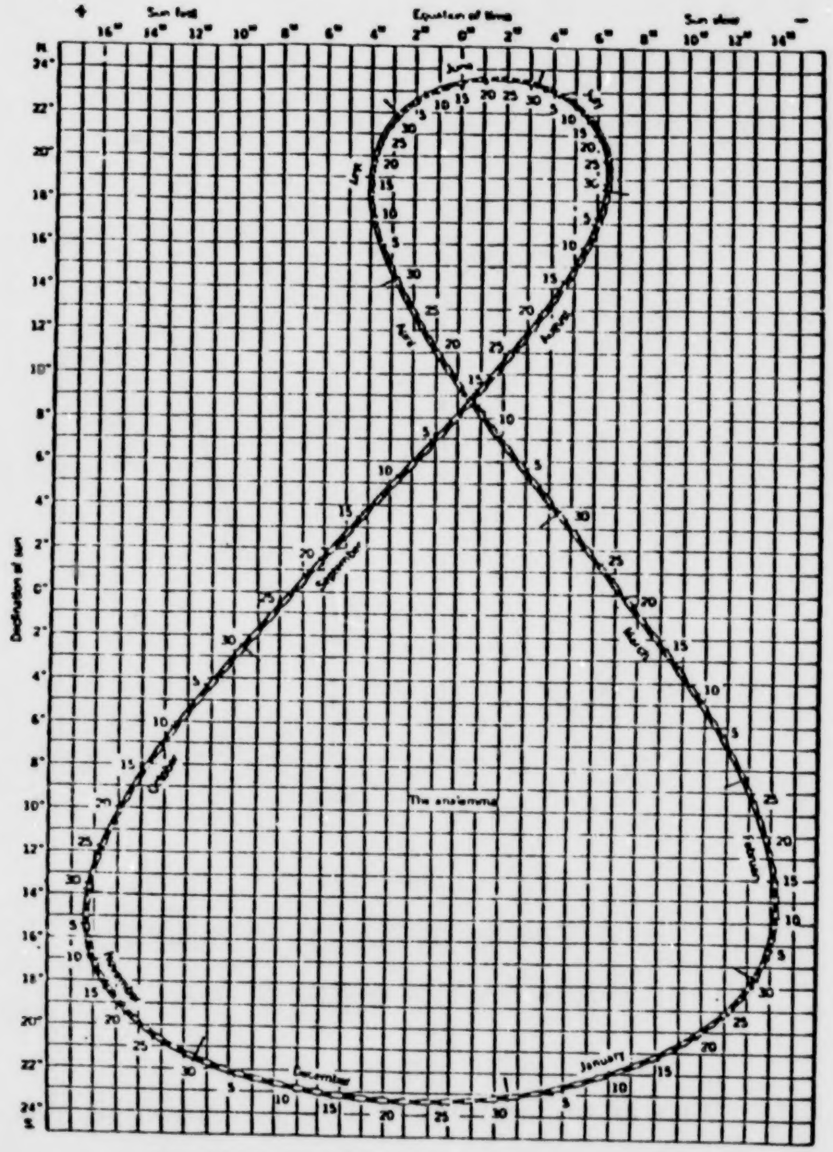


FIGURE 2.3.2

THE EQUATION OF TIME, E, IN MINUTES,

AS A FUNCTION OF TIME OF YEAR

Another form of the equation of time is a graphical presentation known as the analemma. The analemma, Figure 2.3.3, can also be used to determine the declination of the sun.



The analemma—a graphical presentation of the equation of time. Equation of time values are plotted to the right and left of the vertical centerline.

FIGURE 2.3.3

THE ANALEMMA

Figure 2.3.4 and Figure 2.3.5 illustrate the effects of collector tilt on the amount of radiation intercepted.

By applying Equation 2.3.5 and the solar intensity one can find the radiation on a horizontal surface outside the earth's atmosphere on an average day in each month. This is presented in graphical form in Figures 2.3.6 and 2.3.7 and in numerical form in Table 2.3.1.

A study of these curves makes it apparent that the north pole receives more horizontal radiation than any other area of the globe in midsummer, even though the sun is higher further south. This raises the question of how to take advantage of this fact while using a flat-plate collector.

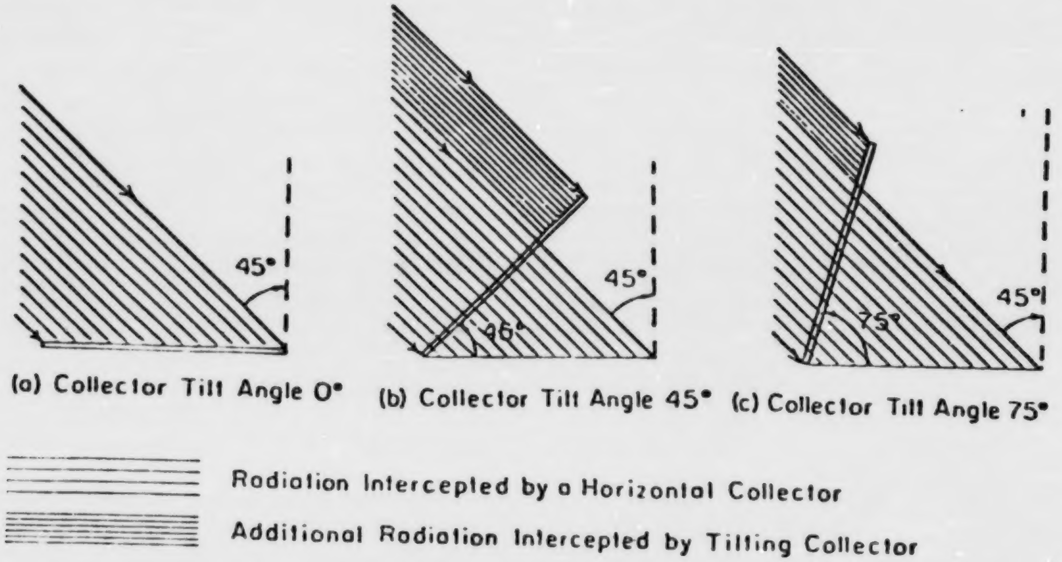
Table 2.3.1 will be of interest later since it contains data which will be needed in handling the problem of eliminating diffuse radiation.

R_b, a geometric factor, is the ratio of the beam radiation on a tilted surface to that on a horizontal surface at the same instant.

$$R_b = \frac{G_{bt}}{G_b} = \frac{\text{Cos } \Theta_T}{\text{Cos } \Theta_z} \qquad \text{(Eqn 2.3.4)}$$

Using Equation 2.3.1,

$$R_b = \frac{\text{Cos }(\phi-\beta) \text{ Cos } \delta \text{ Cos } \omega + \text{Sin }(\phi+\beta) \text{ Sin } \delta}{\text{Cos } \phi \text{ Cos } \delta \text{ Cos } \omega + \text{Sin } \phi \text{ Sin } \delta} \qquad \text{(Eqn 2.3.5)}$$


45°
45°
46°
75°
45°
Radiation Intercepted by a Horizontal Collector
Additional Radiation Intercepted by Tilting Collector

(a) Collector Tilt Angle 0° (b) Collector Tilt Angle 45° (c) Collector Tilt Angle 75°

FIGURE 2.3.4

EFFECT OF TILTING THE COLLECTOR
ON ENERGY INTERCEPTED

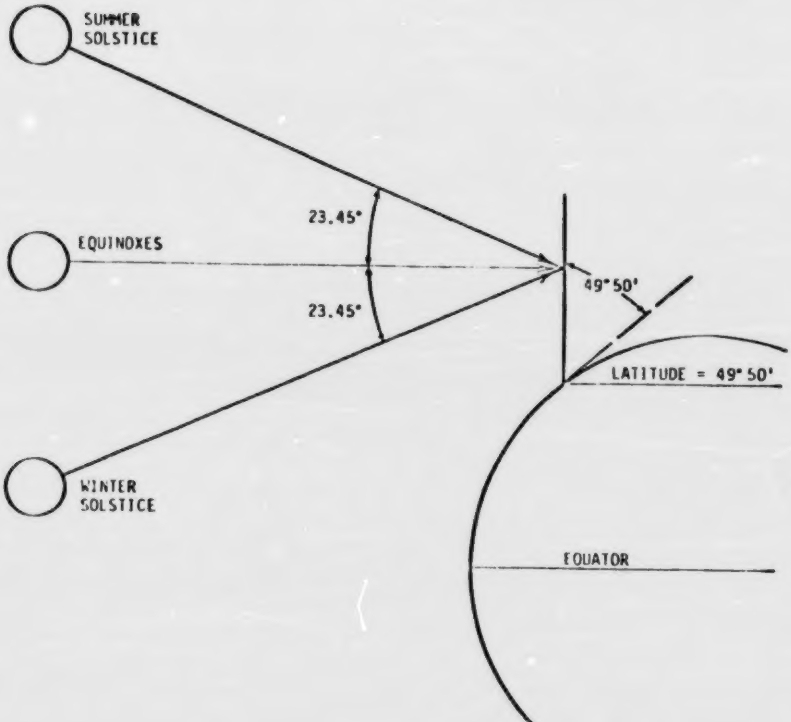
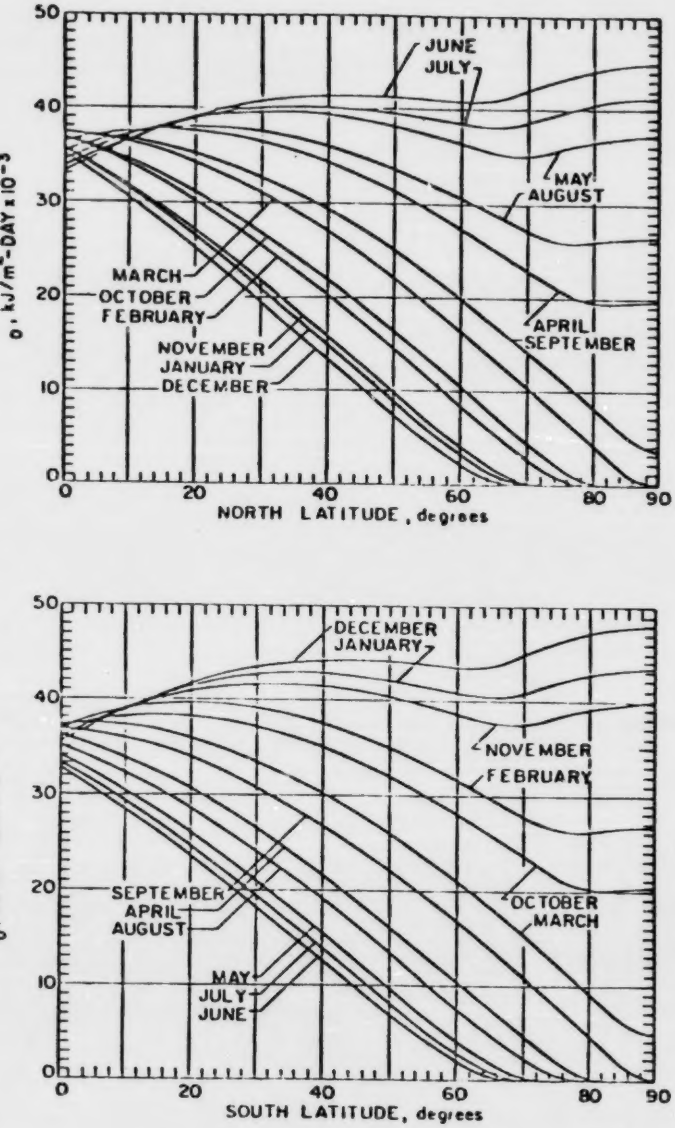

SUMMER
SOLSTICE
EQUINOXES
WINTER
SOLSTICE
23.45°
23.45°
49° 50'
LATITUDE = 49° 50'
EQUATOR

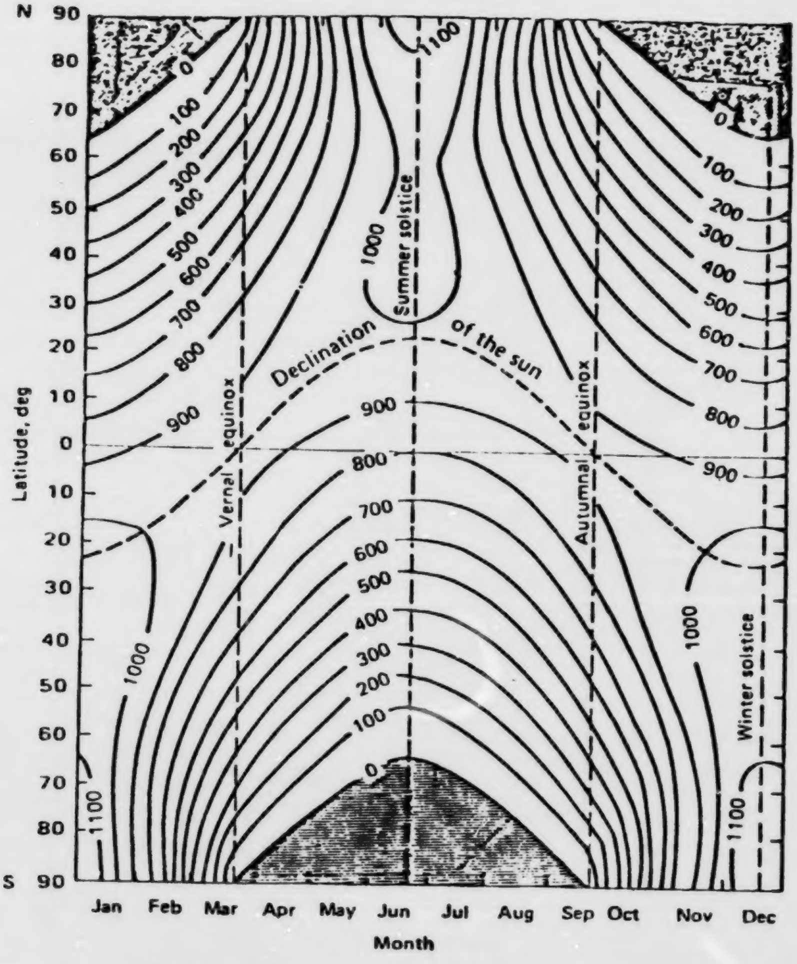
FIGURE 2.3.5

INCIDENT ANGLES ON A COLLECTOR
TILTED AT AN ANGLE EQUAL TO THE LATITUDE



Extraterrestrial daily insolation on a horizontal surface, for mid points of months.

FIGURE 2.3.6



The daily variation of the solar radiation at the top of the atmosphere as a function of latitude. The values are measured in langleys per day. (1 langley = 1 cal/cm² = 3.687 Btu/ft² = 11.6 W · hr/m².) From Kreider, J. F., and F. Kreith, "Solar Heating and Cooling," revised 1st ed., Hemisphere Publ. Corp., Washington, D.C., 1977.

FIGURE 2.3.7

Table 2.3.1

$\overline{I}_0$, Monthly Average Daily Extraterrestrial Radiation on a Horizontal Surface
GJ/m^2 day

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20	26.7	30.4	34.4	37.6	38.9	39.2	39.0	37.9	35.3	31.4	27.6	25.6
25	23.9	28.2	32.9	37.2	39.4	40.1	39.7	37.9	34.3	29.5	24.9	22.7
30	21.1	25.7	31.2	36.5	39.6	40.8	40.1	37.6	32.7	27.3	22.2	19.7
35	18.1	23.1	29.2	35.6	39.6	41.2	40.4	37.0	31.4	24.9	19.3	16.7
40	15.1	20.3	27.1	34.4	39.3	41.4	40.3	36.2	29.6	22.2	16.4	13.6
45	12.0	17.5	24.7	32.9	38.8	41.4	40.1	35.2	27.6	19.6	13.4	10.6
50	8.18	14.5	22.2	31.2	38.1	41.2	39.7	33.9	25.3	16.7	10.4	7.61
55	6.09	11.5	20.1	29.4	36.5	40.9	39.2	32.5	22.9	13.8	7.41	4.80
60	3.40	8.50	16.6	27.3	36.2	40.6	38.6	30.8	20.3	10.8	4.61	2.28

2.4 THE AVAILABILITY OF SOLAR RADIATION DATA

The most detailed solar radiation data available is in the Monthly Radiation Summary of the Atmospheric Environment Service of Environment Canada. This provides hour-by-hour total horizontal radiation for fifty Canadian sites, and at several of these sites also provides diffuse radiation, reflected solar radiation, net radiation or total illumination.

This same data is also available on magnetic tape for those who are using the computer to do hour-by-hour analysis of a solar energy system. This data can be obtained from the Atmospheric Environment Service, Canadian Climate Center, 4905 Dufferin St., Downsview, Ontario, M3H 5T4, Phone (416) 667-4626.

Where no radiation data is available, it is possible to estimate the clear sky radiation by the method of Section 2.8, D and B. Note also that Tables 18 through 26, beginning on Page 27.19 of the 1981 ASHRAE Handbook of Fundamentals provide the direct normal beam radiation for average clear sky conditions for all latitudes and times. Using clear sky radiation, the methods of Section 2.7 and 2.9, from D and B, may be used to estimate the average solar radiation and to give it a reasonable statistical distribution.

Finally, the Solar Programs Office of Public Works Canada provides the monthly average daily total radiation on a horizontal surface for 97 Canadian cities. This data is presented in Appendix 2A at the end of this chapter.

2.5 BEAM AND DIFFUSE COMPONENTS OF MONTHLY AVERAGE RADIATION

Usually, the solar data available (Appendix 2A, for example) is for total horizontal radiation. The geometric factor, R_b, converts the beam radiation on a horizontal surface to that on a tilted surface, but this does not apply to the diffuse component. Using the nomenclature of Figure 2.5.1, the radiation on a horizontal surface is given by

$$G_h = G_b \text{ Cos } \Theta_z + G_d \qquad \text{(Eqn 2.5.1)}$$

and on a tilted surface by

$$G_t = G_b \text{ Cos } \Theta + G_d (1 + \text{Cos } \beta)/2 + G_h \rho_g (1 - \text{Cos } \beta)/2 \qquad \text{(Eqn 2.5.2)}$$

(1+Cosβ)/2 is the view factor from the sky to a tilted collector and (1-Cosβ)/2 is the view factor from the ground.

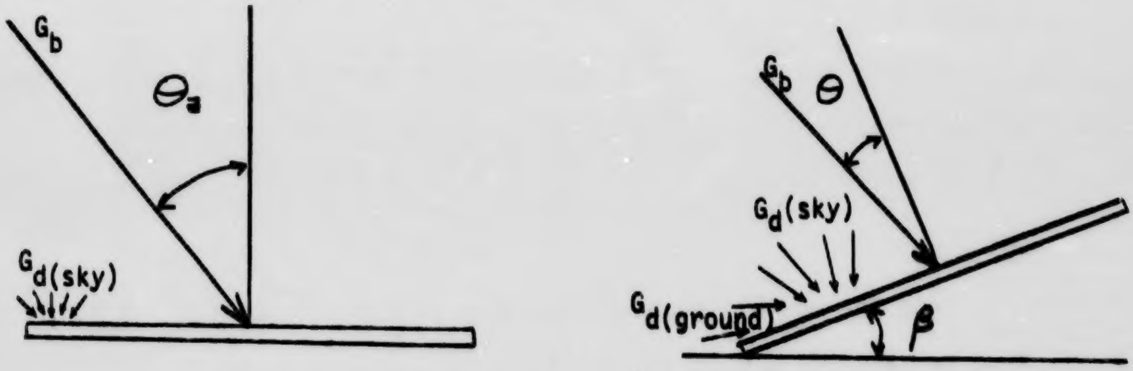


FIGURE 2.5.1

GEOMETRY OF BEAM

AND DIFFUSE RADIATION

The weather data provides G_h, not G_b or G_d, so there is no way of solving these equations and G_d must be estimated from G_h.

There are several methods of making this estimate. One such method is appropriate for dealing with monthly mean daily total radiation, the kind needed as an input to f-Chart. It is based on the use of $\overline{R}$, the ratio of the monthly mean radiation on a tilted surface to the monthly mean radiation on a horizontal surface. This ratio depends on the ratio

$$\overline{K}_T = \overline{H}/\overline{H}_\circ$$

since $\overline{K}_T$ is a measure of the amount of scattering in the atmosphere, and therefore on the ratio of diffuse to direct radiation. Appendix 2A2 at the end of this chapter contains graphs of $\overline{R}$ as a function of $\overline{K}_T$.

Example 1

Find $\overline{H}_T$ for Winnipeg in January for a collector in a vertical position facing south.

Solution

From the monthly mean weather data of Appendix 2A-1, $\overline{H}$ = 5 MJ/m² day.
From Table 2.3.1. $\overline{H}_\circ$ = 719 BTU/ft² day at latitude 50°

$$\overline{H}_\circ = 719 \times .011357 = 8.17 \text{ MJ/m}^2 \text{ day}$$

$$\therefore\ \overline{K}_T = \frac{\overline{H}}{\overline{H}_\circ} = \frac{5}{8.17} = .61$$

From Graph 2A.2 - 27 $\overline{R}$ = 2.95

$$\text{Therefore } \overline{H}_T = \overline{R}\ \overline{H}$$

$$= 2.95 \times 5$$

$$= 14.75 \text{ MJ/m}^2 \text{ day}$$

Example 2

Find $\overline{H}_T$ for a collector tilted at 45° in Vancouver in June.

Solution

From the monthly mean weather data $\overline{H} = 22$ MJ/m² day. From Table 2.3.1. $\overline{H}_o = 3621$ BTU/ft² day

$$\therefore \overline{H}_o = 3621 \times .011357$$

$$= 41.1 \text{ MJ/m}^2 \text{ day}$$

and $\overline{K}_T = .53$

Interpolating between Figures 5,6,12 and 13 in Appendix 2A.2,

$$\overline{R} = .88$$

$$\therefore \overline{H}_T = \overline{R}\ \overline{H}$$

$$= .88 \times 22$$

$$= 19.36 \text{ MJ/m}^2 \text{ day}$$

Assignment

1. Find the mean monthly value of the daily solar radiation on a south facing collector at 43° 41' in Toronto in November.

2. Find $\overline{H}_T$ for Medicine Hat for a south facing collector at 35° in July. $\overline{H} = 25$ MJ/m^2 day.

APPENDIX 2A-1

MONTHLY MEAN WEATHER DATA
FOR CANADIAN CITIES

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 ............ CANADIAN CITY DATA USED IN F-CHART ..................

                                                 JAN  FEB  MAR  APR  MAY  JUN  JUL  AUG  SEP  OCT  NOV  DEC
  1 ABBOTSFORD      BC   D-D (BASED ON 18 C)     515  393  334 [illegible] [illegible] 99 52  52  110  245  370  463
   (LATITUDE= 49.02)     MEAN AMB. TEMP. (C)       1    4    6    9   12   15   17   17   14   10    6    3
                         SOLAR (MJ/M**2/DAY)       3    6   11   16   21   23   23   19   14    7    4    2

  2 AKLAVIK         NWT  D-D (BASED ON 18 C)    1452 1299 1265  930  591  269  152  255  448  796 1147 1496
   (LATITUDE= 68.22)     MEAN AMB. TEMP. (C)     -28  -27  -22  -12    0    9   13   10    3   -7  -19  -26
                         SOLAR (MJ/M**2/DAY)       0    2    8   16   20   22   19   12    6    2    0    0

  3 ARMSTRONG       ONT  D-D (BASED ON 18 C)    1201 1018  893  556  259  161   75  118  267  447  736 1051
   (LATITUDE= 50.28)     MEAN AMB. TEMP. (C)     -21  -18  -11   -1    6   13   16   15    9    4   -7  -16
                         SOLAR (MJ/M**2/DAY)       5    9   14   18   20   22   22   18   12    7    5    3

  4 BAGOTVILLE      QUE  D-D (BASED ON 18 C)    1046  893  762  485  283  100   45   74  198  382  587  921
   (LATITUDE= 48.33)     MEAN AMB. TEMP. (C)     -16  -14   -7    2    9   15   18   17   12    6   -2  -12
                         SOLAR (MJ/M**2/DAY)       5    9   15   17   19   21   20   17   12    8    4    4

  5 BAIE COMEAU     QUE  D-D (BASED ON 18 C)     974  888  762  553  371  157   87   75  233  409  589  841
   (LATITUDE= 49.23)     MEAN AMB. TEMP. (C)     -14  -13   -7    0    7   13   16   15   10    5   -2  -10
                         SOLAR (MJ/M**2/DAY)       4    8   14   17   18   20   19   16   12    7    4    3

  6 BANFF           ALTA D-D (BASED ON 18 C)     906  693  675  470  327  207  114  145  268  427  653  834
   (LATITUDE= 51.18)     MEAN AMB. TEMP. (C)     -11   -7   -4    2    8   11   15   13    9    4   -4   -8
                         SOLAR (MJ/M**2/DAY)       4    8   13   17   21   23   25   20   15    9    5    3

  7 BLUE RIVER      BC   D-D (BASED ON 18 C)     896  656  603  432  259  129   68   72  223  415  615  793
   (LATITUDE= 52.12)     MEAN AMB. TEMP. (C)     -11   -4   -1    4   10   14   16   15   11    6   -2   -7
                         SOLAR (MJ/M**2/DAY)       4    7   12   17   21   23   23   19   16    7    4    3

  8 BRANDON         MAN  D-D (BASED ON 18 C)    1144  934  860  453  254   86   24   46  195  382  699  989
   (LATITUDE= 49.58)     MEAN AMB. TEMP. (C)     -19  -15   -9    3   10   16   19   18   12    6   -5  -14
                         SOLAR (MJ/M**2/DAY)       5    9   14   18   21   23   24   20   14    9    5    4

  9 CALGARY INT'L   ALTA D-D (BASED ON 18 C)     899  719  691  439  269  150   65   97  221  383  619  793
   (LATITUDE= 51.10)     MEAN AMB. TEMP. (C)     -11   -7   -4    3    9   13   17   15   11    6   -3   -8
                         SOLAR (MJ/M**2/DAY)       4    8   13   17   21   23   25   20   15    9    5    3

 10 CASTLEGAR       BC   D-D (BASED ON 18 C)     681  526  468  307  151   58   17   17  106  323  472  621
   (LATITUDE= 49.33)     MEAN AMB. TEMP. (C)      -4   -1    3    8   13   17   21   20   15    8    2   -2
                         SOLAR (MJ/M**2/DAY)       4    7   12   17   21   23   25   21   15    9    5    3

 11 CHARLO          NB   D-D (BASED ON 18 C)     899  795  695  471  285  107   31   54  172  351  530  791
   (LATITUDE= 48.00)     MEAN AMB. TEMP. (C)     -12  -11   -6    1    7   13   17   16   11    5   -1   -8
                         SOLAR (MJ/M**2/DAY)       5    8   13   16   19   20   20   17   13    8    4    4

 12 CHARLOTTETOWN   PEI  D-D (BASED ON 18 C)     767  714  657  470  292  126   27   37  130  290  441  672
   (LATITUDE= 46.28)     MEAN AMB. TEMP. (C)      -7   -7   -3    2    9   14   18   18   14    9    3   -4
                         SOLAR (MJ/M**2/DAY)       5    9   13   15   18   21   21   18   13    8    5    4

 13 CHATHAM         NB   D-D (BASED ON 18 C)     848  755  663  451  266  101   24   43  148  325  502  758
   (LATITUDE= 47.03)     MEAN AMB. TEMP. (C)      -9   -9   -3    3   10   15   19   18   13    8    1   -7
                         SOLAR (MJ/M**2/DAY)       5    8   13   16   19   20   20   17   14    8    5    4

 14 CHIBOUGAMAU     QUE  D-D (BASED ON 18 C)    1181 1045  909  606  362  121   59  112  294  477  694 1054
   (LATITUDE= 49.50)     MEAN AMB. TEMP. (C)     -20  -18  -10   -1    6   13   16   15   10    4   -4  -15
                         SOLAR (MJ/M**2/DAY)       5    9   15   17   19   21   19   14   12    7    4    3
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·················· CANADIAN CITY DATA USED IN FCHART ··················

				JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
15	CHURCHILL	MAN	D-D (BASED ON 18 C)	1414	1260	1177	871	620	359	194	204	366	590	902	1234
	(LATITUDE= 58.78)		MEAN AMB. TEMP. (C)	-26	-27	-20	-11	-2	6	12	12	6	-1	-12	-22
			SOLAR (MJ/M**2/DAY)	2	6	13	19	21	22	21	16	9	4	2	1
16	COMOX	BC	D-D (BASED ON 18 C)	494	398	404	302	185	93	43	51	129	276	389	448
	(LATITUDE= 49.72)		MEAN AMB. TEMP. (C)	2	4	5	8	12	15	17	17	14	9	5	4
			SOLAR (MJ/M**2/DAY)	3	6	10	16	20	22	23	18	13	7	4	2
17	CORONATION	ALTA	D-D (BASED ON 18 C)	1061	854	779	463	247	124	49	80	221	410	696	922
	(LATITUDE= 52.10)		MEAN AMB. TEMP. (C)	-16	-12	-7	3	10	14	17	16	11	4	-5	-12
			SOLAR (MJ/M**2/DAY)	4	8	14	18	21	23	24	19	14	9	5	3
18	CRANBROOK	BC	D-D (BASED ON 18 C)	829	641	577	367	227	118	41	61	185	378	590	748
	(LATITUDE= 49.50)		MEAN AMB. TEMP. (C)	-9	-5	-1	6	11	14	18	17	12	5	-2	-10
			SOLAR (MJ/M**2/DAY)	4	8	13	17	21	23	25	21	15	9	5	3
19	CREE LAKE	SASK	D-D (BASED ON 18 C)	1351	1092	987	581	327	128	83	117	333	536	877	1215
	(LATITUDE= 57.30)		MEAN AMB. TEMP. (C)	-24	-20	-12	-2	6	13	16	15	8	2	-11	-20
			SOLAR (MJ/M**2/DAY)	3	6	10	19	23	23	22	17	10	6	3	2
20	DAUPHIN	MAN	D-D (BASED ON 18 C)	1159	950	849	485	261	98	32	54	198	370	693	1001
	(LATITUDE= 51.15)		MEAN AMB. TEMP. (C)	-19	-16	-9	2	10	15	19	17	12	6	-5	-14
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	22	23	19	13	8	4	4
21	DAWSON	YUK	D-D (BASED ON 18 C)	1443	1152	996	593	318	127	81	166	347	659	1039	1352
	(LATITUDE= 64.07)		MEAN AMB. TEMP. (C)	-29	-23	-14	-2	8	14	16	13	6	-3	-17	-25
			SOLAR (MJ/M**2/DAY)	1	3	9	15	19	21	19	14	7	3	1	0
22	EARLTON	ONT	D-D (BASED ON 18 C)	1058	897	792	493	274	106	52	94	204	375	607	934
	(LATITUDE= 47.70)		MEAN AMB. TEMP. (C)	-16	-14	-8	2	9	15	18	16	11	6	-2	-12
			SOLAR (MJ/M**2/DAY)	5	9	15	17	20	22	21	18	12	8	4	4
23	EDMONTON	ALTA	D-D (BASED ON 18 C)	1016	806	727	421	222	109	47	62	215	398	666	890
	(LATITUDE= 53.57)		MEAN AMB. TEMP. (C)	-15	-11	-5	4	11	15	19	16	11	5	-4	-11
			SOLAR (MJ/M**2/DAY)	4	7	13	18	20	23	23	18	13	8	4	3
24	ESTEVAN	SASK	D-D (BASED ON 18 C)	1050	854	750	423	228	84	20	37	177	355	653	911
	(LATITUDE= 49.20)		MEAN AMB. TEMP. (C)	-16	-13	-6	4	11	16	20	19	13	7	-4	-12
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	25	20	15	10	5	4
25	FORT MCMURRAY	ALTA	D-D (BASED ON 18 C)	1237	978	850	497	279	141	69	112	271	463	795	1086
	(LATITUDE= 56.65)		MEAN AMB. TEMP. (C)	-22	-17	-9	1	9	14	16	15	9	3	-8	-17
			SOLAR (MJ/M**2/DAY)	3	6	12	19	21	23	22	18	11	6	3	2
26	FORT NELSON	BC	D-D (BASED ON 18 C)	1276	994	843	505	259	112	58	107	275	523	910	1201
	(LATITUDE= 58.83)		MEAN AMB. TEMP. (C)	-23	-17	-9	1	10	15	17	15	9	1	-12	-21
			SOLAR (MJ/M**2/DAY)	2	5	10	16	20	20	20	16	10	5	2	1
27	FORT SMITH	NWT	D-D (BASED ON 18 C)	1403	1145	1010	637	336	153	79	124	310	542	897	1216
	(LATITUDE= 60.00)		MEAN AMB. TEMP. (C)	-27	-23	-15	-3	7	13	16	14	8	1	-12	-21
			SOLAR (MJ/M**2/DAY)	2	5	10	17	21	21	21	16	10	4	2	1
28	FORT ST. JOHN	BC	D-D (BASED ON 18 C)	1093	846	758	465	254	136	77	112	242	430	736	970
	(LATITUDE= 56.25)		MEAN AMB. TEMP. (C)	-17	-12	-6	3	10	14.	16	15	10	4	-7	-13
			SOLAR (MJ/M**2/DAY)	3	6	12	16	20	23	21	17	11	6	3	2

[illegible]

Station		JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
29 FREDERICTON NB	D-D (BASED ON 18 C)	[illegible]	727	615	418	225	68	21	25	147	315	489	[illegible]
(LATITUDE= 45.87)	MEAN AMB. TEMP. (C)	-9	-8	-2	4	11	16	19	18	13	8	2	-6
	SOLAR (MJ/M**2/DAY)	5	9	13	15	18	21	20	17	13	9	5	4
30 FROBISHER BAY NWT	D-D (BASED ON 18 C)	1369	1236	1254	962	660	434	312	342	467	706	912	1191
(LATITUDE= 63.75)	MEAN AMB. TEMP. (C)	-26	-25	-22	-14	-3	4	8	7	2	-5	-12	-20
	SOLAR (MJ/M**2/DAY)	1	4	10	17	23	21	16	13	6	3	1	0
31 GANDER INT'L NFLD	D-D (BASED ON 18 C)	747	687	669	516	362	202	73	84	189	362	483	665
(LATITUDE= 48.95)	MEAN AMB. TEMP. (C)	-6	-6	-4	1	6	11	17	16	12	6	2	-3
	SOLAR (MJ/M**2/DAY)	4	7	11	14	17	18	18	16	12	7	4	3
32 GOOSE BAY NFLD	D-D (BASED ON 18 C)	1065	914	821	594	407	214	91	118	248	458	650	942
(LATITUDE= 53.32)	MEAN AMB. TEMP. (C)	-16	-14	-8	-2	5	11	16	15	10	3	-4	-12
	SOLAR (MJ/M**2/DAY)	3	7	12	16	18	18	18	14	10	6	3	3
33 GRANDE PRAIRIE ALTA	D-D (BASED ON 18 C)	1095	861	790	469	249	133	75	107	236	431	731	969
(LATITUDE= 55.17)	MEAN AMB. TEMP. (C)	-17	-12	-7	2	10	14	16	15	10	4	-6	-13
	SOLAR (MJ/M**2/DAY)	4	6	10	16	20	27	21	21	13	7	3	2
34 HALIFAX NS	D-D (BASED ON 18 C)	676	628	582	419	278	133	37	30	105	251	391	593
(LATITUDE= 44.65)	MEAN AMB. TEMP. (C)	-3	-3	0	5	10	14	18	19	16	11	6	-1
	SOLAR (MJ/M**2/DAY)	5	8	12	15	17	20	19	18	14	9	5	4
35 INUVIK NWT	D-D (BASED ON 18 C)	1506	1337	1320	966	594	244	147	231	458	801	1189	1291
(LATITUDE= 68.30)	MEAN AMB. TEMP. (C)	-29	-29	-24	-15	-1	10	13	10	3	-7	-21	-27
	SOLAR (MJ/M**2/DAY)	0	2	8	16	20	22	18	12	6	2	1	0
36 JASPER ALTA	D-D (BASED ON 18 C)	936	694	643	440	288	166	96	126	242	411	654	836
(LATITUDE= 52.88)	MEAN AMB. TEMP. (C)	-12	-7	-3	3	9	13	15	14	10	5	-4	-9
	SOLAR (MJ/M**2/DAY)	3	7	12	17	21	23	23	19	15	7	4	3
37 KAMLOOPS BC	D-D (BASED ON 18 C)	742	543	458	272	128	41	11	20	106	298	495	642
(LATITUDE= 50.68)	MEAN AMB. TEMP. (C)	-6	-1	4	9	14	18	21	20	15	8	2	-3
	SOLAR (MJ/M**2/DAY)	3	6	12	17	21	23	23	19	14	7	4	3
38 KAPUSKASING ONT	D-D (BASED ON 18 C)	1123	956	841	521	319	136	70	105	232	401	667	995
(LATITUDE= 49.42)	MEAN AMB. TEMP. (C)	-18	-16	-9	1	8	14	17	15	10	5	-4	-14
	SOLAR (MJ/M**2/DAY)	5	8	13	16	18	22	20	17	12	7	4	3
39 KENORA ONT	D-D (BASED ON 18 C)	1109	924	776	465	259	91	27	47	196	373	678	996
(LATITUDE= 49.80)	MEAN AMB. TEMP. (C)	-18	-15	-7	3	10	16	19	18	12	6	-5	-14
	SOLAR (MJ/M**2/DAY)	5	9	14	18	20	23	23	19	13	8	5	4
40 KINGSTON ONT	D-D (BASED ON 18 C)	793	702	612	377	202	57	9	17	87	256	439	711
(LATITUDE= 44.25)	MEAN AMB. TEMP. (C)	-7	-8	-2	5	12	17	21	20	16	10	4	-3
	SOLAR (MJ/M**2/DAY)	5	8	13	16	20	22	22	19	15	9	5	4
41 LETHBRIDGE ALTA	D-D (BASED ON 18 C)	851	666	634	383	221	106	29	48	168	323	560	729
(LATITUDE= 49.70)	MEAN AMB. TEMP. (C)	-9	-6	-2	5	11	15	19	18	13	8	-1	-6
	SOLAR (MJ/M**2/DAY)	5	8	14	18	22	24	25	21	15	10	6	4
42 LONDON ONT	D-D (BASED ON 18 C)	746	668	582	342	183	48	12	22	98	251	446	670
(LATITUDE= 42.98)	MEAN AMB. TEMP. (C)	-6	-6	-1	7	12	18	21	20	16	10	3	-4
	SOLAR (MJ/M**2/DAY)	5	8	13	16	21	23	23	19	15	10	5	5

.... CANADIAN CITY DATA USED IN F-CHART ....

				JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
43	MANIWAKI	QUE	D-D (BASED ON 18 C)	975	841	722	426	240	57	40	67	176	346	528	868
	(LATITUDE= 46.38)		MEAN AMB. TEMP. (C)	-13	-12	-5	4	10	16	18	17	12	7	0	-10
			SOLAR (MJ/M**2/DAY)	5	9	14	16	20	22	21	18	13	9	5	4
44	MEDICINE HAT	ALTA	D-D (BASED ON 18 C)	934	739	659	361	189	79	17	34	157	324	589	792
	(LATITUDE= 50.02)		MEAN AMB. TEMP. (C)	-12	-8	-3	6	12	16	20	19	13	8	-2	-8
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	25	21	15	10	5	4
45	MONCTON	NB	D-D (BASED ON 18 C)	806	722	647	447	261	109	26	44	145	298	472	722
	(LATITUDE= 46.10)		MEAN AMB. TEMP. (C)	-8	-8	-3	4	10	15	19	18	13	8	2	-5
			SOLAR (MJ/M**2/DAY)	4	8	12	16	18	19	20	17	13	9	4	4
46	MONT JOLI	QUE	D-D (BASED ON 18 C)	900	798	713	487	305	129	57	77	191	363	538	795
	(LATITUDE= 48.60)		MEAN AMB. TEMP. (C)	-11	-10	-5	2	8	14	17	16	12	6	0	-8
			SOLAR (MJ/M**2/DAY)	5	8	14	17	19	20	20	17	12	8	4	4
47	MONTREAL INT'L	QUE	D-D (BASED ON 18 C)	865	757	636	369	168	38	7	20	106	269	471	765
	(LATITUDE= 45.50)		MEAN AMB. TEMP. (C)	-10	-9	-2	6	13	19	21	20	15	9	2	-7
			SOLAR (MJ/M**2/DAY)	5	9	13	16	19	21	21	18	14	9	5	4
48	MOOSONEE	ONT	D-D (BASED ON 18 C)	1163	1020	928	608	392	200	106	130	244	414	674	1032
	(LATITUDE= 51.27)		MEAN AMB. TEMP. (C)	-20	-18	-12	-2	5	12	16	14	10	5	-4	-15
			SOLAR (MJ/M**2/DAY)	4	8	13	17	18	20	18	15	11	6	3	3
49	MOUNT FOREST	ONT	D-D (BASED ON 18 C)	822	755	654	406	245	91	41	61	140	299	490	745
	(LATITUDE= 43.97)		MEAN AMB. TEMP. (C)	-9	-9	-3	5	10	16	18	17	14	8	2	-6
			SOLAR (MJ/M**2/DAY)	5	8	12	16	20	23	23	19	15	10	5	4
50	MUSKOKA	ONT	D-D (BASED ON 18 C)	869	762	674	401	220	80	33	51	150	306	505	775
	(LATITUDE= 44.97)		MEAN AMB. TEMP. (C)	-10	-9	-4	5	11	16	19	18	14	8	1	-7
			SOLAR (MJ/M**2/DAY)	5	8	13	16	21	23	23	19	14	9	5	4
51	NATASHQUAN	QUE	D-D (BASED ON 18 C)	927	823	746	560	414	239	125	145	259	419	563	832
	(LATITUDE= 50.20)		MEAN AMB. TEMP. (C)	-12	-11	-6	-1	5	10	14	13	9	5	-1	-9
			SOLAR (MJ/M**2/DAY)	4	7	12	16	18	19	19	16	12	7	4	3
52	NORTH BAY	ONT	D-D (BASED ON 18 C)	956	823	725	444	249	87	37	62	176	343	564	853
	(LATITUDE= 46.32)		MEAN AMB. TEMP. (C)	-13	-11	-5	3	10	16	18	17	12	7	-1	-9
			SOLAR (MJ/M**2/DAY)	5	9	14	17	20	23	22	19	13	9	5	4
53	OTTAWA INT'L	ONT	D-D (BASED ON 18 C)	899	778	654	371	181	44	10	28	123	290	499	796
	(LATITUDE= 45.32)		MEAN AMB. TEMP. (C)	-11	-10	-3	6	12	18	21	19	15	9	1	-8
			SOLAR (MJ/M**2/DAY)	6	9	14	15	20	22	22	18	14	9	5	4
54	PEACE RIVER	ALTA	D-D (BASED ON 18 C)	1163	881	803	468	267	130	71	111	263	447	800	1020
	(LATITUDE= 56.25)		MEAN AMB. TEMP. (C)	-21	-14	-9	2	9	14	16	14	9	4	-9	-15
			SOLAR (MJ/M**2/DAY)	3	6	12	18	20	20	22	18	12	7	3	2
55	PENTICTON	BC	D-D (BASED ON 18 C)	649	499	445	278	147	52	13	19	107	287	448	570
	(LATITUDE= 49.47)		MEAN AMB. TEMP. (C)	-3	0	4	9	13	17	20	19	15	9	3	0
			SOLAR (MJ/M**2/DAY)	3	7	12	17	21	23	24	20	15	8	4	3
56	PORT HARDY	BC	D-D (BASED ON 18 C)	494	401	421	345	262	186	136	131	183	289	390	451
	(LATITUDE= 50.67)		MEAN AMB. TEMP. (C)	2	4	4	7	10	12	14	14	12	9	5	3
			SOLAR (MJ/M**2/DAY)	3	6	9	13	18	19	20	15	11	6	3	2

.... CANADIAN CITY DATA USED IN F-CHART . ...

				JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
57	POSTE DE LA BA	QUE	D-D (BASED ON 18 C)	1224	1145	1056	752	521	362	209	251	317	491	682	1037
	(LATITUDE= 55.28)		MEAN AMB. TEMP. (C)	-22	-22	-17	-7	1	6	11	10	8	2	-5	-15
			SOLAR (MJ/M**2/DAY)	3	7	12	17	18	20	18	14	9	5	3	2
58	PRINCE ALBERT	SASK	D-D (BASED ON 18 C)	1204	987	863	489	265	119	45	74	234	436	752	1073
	(LATITUDE= 53.17)		MEAN AMB. TEMP. (C)	-21	-17	-10	2	10	14	18	16	10	4	-7	-16
			SOLAR (MJ/M**2/DAY)	4	8	14	19	21	22	23	19	13	8	4	3
59	PRINCE GEORGE	BC	D-D (BASED ON 18 C)	925	685	623	422	265	152	102	135	245	413	626	795
	(LATITUDE= 53.83)		MEAN AMB. TEMP. (C)	-12	-6	-2	4	9	13	15	14	10	5	-3	-8
			SOLAR (MJ/M**2/DAY)	3	6	11	16	19	22	21	18	12	6	3	2
60	PRINCE RUPERT	BC	D-D (BASED ON 18 C)	578	411	455	390	309	213	168	157	193	303	418	522
	(LATITUDE= 54.28)		MEAN AMB. TEMP. (C)	0	3	3	5	9	11	13	13	12	8	4	1
			SOLAR (MJ/M**2/DAY)	3	5	10	14	18	19	17	14	9	5	3	1
61	QUEBEC	QUE	D-D (BASED ON 18 C)	919	810	692	441	232	76	22	45	155	332	533	823
	(LATITUDE= 46.81)		MEAN AMB. TEMP. (C)	-12	-11	-4	3	11	16	19	18	13	7	0	-9
			SOLAR (MJ/M**2/DAY)	5	9	14	17	19	21	21	17	13	8	5	4
62	RED DEER	ALTA	D-D (BASED ON 18 C)	1021	817	751	443	260	143	69	106	228	416	679	902
	(LATITUDE= 52.27)		MEAN AMB. TEMP. (C)	-15	-11	-6	3	10	13	16	15	10	5	-5	-11
			SOLAR (MJ/M**2/DAY)	4	8	13	18	21	23	22	17	14	9	5	3
63	REGINA	SASK	D-D (BASED ON 18 C)	1094	914	817	441	234	97	29	46	198	394	697	959
	(LATITUDE= 50.45)		MEAN AMB. TEMP. (C)	-17	-14	-8	3	11	15	19	18	12	5	-5	-13
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	24	20	14	9	5	4
64	RESOLUTE	NWT	D-D (BASED ON 18 C)	1567	1456	1527	1232	889	549	425	474	687	1016	1275	1452
	(LATITUDE= 74.68)		MEAN AMB. TEMP. (C)	-33	-34	-31	-23	-11	0	4	3	-5	-15	-24	-29
			SOLAR (MJ/M**2/DAY)	0	1	5	15	23	25	18	11	5	1	0	0
65	REVELSTOKE	BC	D-D (BASED ON 18 C)	742	561	500	323	163	68	21	36	130	244	516	669
	(LATITUDE= 51.00)		MEAN AMB. TEMP. (C)	-7	-3	1	6	13	16	19	17	13	7	1	-4
			SOLAR (MJ/M**2/DAY)	4	7	12	17	21	23	24	20	14	8	4	4
66	RIVIERE AU REN	QUE	D-D (BASED ON 18 C)	849	788	717	523	351	160	77	92	201	371	520	738
	(LATITUDE= 49.02)		MEAN AMB. TEMP. (C)	-10	-9	-5	0	6	12	16	16	12	6	1	-6
			SOLAR (MJ/M**2/DAY)	4	5	10	16	18	17	19	16	12	7	4	3
67	RIVIER DU LOUP	QUE	D-D (BASED ON 18 C)	942	833	728	507	305	108	42	83	202	386	568	833
	(LATITUDE= 47.80)		MEAN AMB. TEMP. (C)	-12	-12	-6	1	8	14	17	16	12	6	-1	-9
			SOLAR (MJ/M**2/DAY)	5	9	14	17	19	21	20	17	13	8	4	4
68	SAINT JOHN	NB	D-D (BASED ON 18 C)	790	721	640	447	286	135	49	62	162	313	464	702
	(LATITUDE= 45.27)		MEAN AMB. TEMP. (C)	-7	-7	-1	4	10	14	17	17	14	9	3	-4
			SOLAR (MJ/M**2/DAY)	5	9	13	16	19	21	20	18	14	9	5	4
69	SASKATOON	SASK	D-D (BASED ON 18 C)	1139	936	829	445	233	97	30	54	204	402	714	994
	(LATITUDE= 52.23)		MEAN AMB. TEMP. (C)	-19	-15	-9	3	11	15	19	17	11	5	-6	-14
			SOLAR (MJ/M**2/DAY)	4	8	13	17	21	22	24	19	14	8	4	3
70	SCHEFFERVILLE	QUE	D-D (BASED ON 18 C)	1261	1106	1016	750	529	287	172	227	372	598	803	1118
	(LATITUDE= 54.80)		MEAN AMB. TEMP. (C)	-23	-21	-15	-7	1	9	13	11	6	-1	-9	-18
			SOLAR (MJ/M**2/DAY)	3	7	13	17	18	19	17	15	10	6	3	2

******************** CANADIAN CITY DATA USED IN FC-HAT ********************

No.	Station	Prov.	Data	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
71	SEPT-ILLES	QUE	D-D (BASED ON 18 C)	989	867	767	549	351	194	95	122	252	436	607	875
	(LATITUDE= 50.22)		MEAN AMB. TEMP. (C)	-14	-13	-7	0	6	12	15	14	10	4	-2	-10
			SOLAR (MJ/M**2/DAY)	5	9	12	14	19	20	18	16	12	8	4	3
72	SHERBROOKE	QUE	D-D (BASED ON 18 C)	918	825	697	438	249	91	49	76	188	337	536	828
	(LATITUDE= 45.40)		MEAN AMB. TEMP. (C)	-12	-11	-4	4	10	15	18	17	12	7	1	-8
			SOLAR (MJ/M**2/DAY)	5	9	13	16	19	22	21	18	14	9	5	4
73	SIMCO	ONT	D-D (BASED ON 18 C)	731	662	567	343	176	44	12	22	98	246	414	647
	(LATITUDE= 42.85)		MEAN AMB. TEMP. (C)	-6	-6	0	7	12	18	20	19	16	10	4	-3
			SOLAR (MJ/M**2/DAY)	5	8	13	16	20	23	22	19	15	10	5	4
74	ST. JOHN'S	NFLD	D-D (BASED ON 18 C)	674	629	634	507	387	229	98	91	184	337	425	599
	(LATITUDE= 47.57)		MEAN AMB. TEMP. (C)	-4	-4	-2	1	6	10	15	15	12	7	4	-1
			SOLAR (MJ/M**2/DAY)	3	6	10	13	17	18	18	14	12	7	3	3
75.	STEPHENVILLE	NFLD	D-D (BASED ON 18 C)	707	674	646	490	346	197	75	67	172	328	447	634
	(LATITUDE= 48.53)		MEAN AMB. TEMP. (C)	-5	-6	-3	2	7	12	16	16	12	7	3	-2
			SOLAR (MJ/M**2/DAY)	4	7	12	14	18	19	19	16	12	8	4	3
76	SUDBURY	ONT	D-D (BASED ON 18 C)	982	860	752	452	255	87	33	59	176	348	572	871
	(LATITUDE= 46.48)		MEAN AMB. TEMP. (C)	-13	-12	-6	3	10	16	19	18	13	7	-1	-10
			SOLAR (MJ/M**2/DAY)	5	9	14	17	21	23	23	19	13	8	4	4
77	SWIFT CURRENT	SASK	D-D (BASED ON 18 C)	991	809	742	429	242	110	33	54	194	370	648	920
	(LATITUDE= 50.33)		MEAN AMB. TEMP. (C)	-14	-11	-6	4	10	15	19	18	12	6	-4	-10
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	25	20	14	10	5	4
78	SYDNEY	NS	D-D (BASED ON 18 C)	696	664	633	475	320	158	41	41	130	293	413	605
	(LATITUDE= 46.17)		MEAN AMB. TEMP. (C)	-4	-6	-2	2	8	13	18	18	14	9	4	-2
			SOLAR (MJ/M**2/DAY)	4	8	12	14	19	20	20	17	13	8	4	3
79	TERRACE	BC	D-D (BASED ON 18 C)	734	536	504	369	244	130	77	90	184	365	534	663
	(LATITUDE= 54.52)		MEAN AMB. TEMP. (C)	-6	-1	2	6	10	14	16	16	12	6	0	-3
			SOLAR (MJ/M**2/DAY)	2	5	10	14	18	19	18	15	9	5	3	2
80	THE PAS	MAN	D-D (BASED ON 18 C)	1253	1026	913	554	322	130	40	74	237	439	766	1098
	(LATITUDE= 53.82)		MEAN AMB. TEMP. (C)	-22	-18	-11	0	8	14	18	16	10	4	-8	-17
			SOLAR (MJ/M**2/DAY)	3	7	13	18	20	20	21	17	11	6	3	3
81	THOMPSON	MAN	D-D (BASED ON 18 C)	1381	1136	993	619	388	178	97	137	329	573	874	1225
	(LATITUDE= 55.80)		MEAN AMB. TEMP. (C)	-26	-22	-14	-3	6	12	15	14	7	0	-11	-22.
			SOLAR (MJ/M**2/DAY)	3	7	13	19	22	22	22	16	10	6	3	2
82	THUNDER BAY	ONT	D-D (BASED ON 18 C)	1017	877	749	469	301	131	47	71	204	371	616	893
	(LATITUDE= 48.37)		MEAN AMB. TEMP. (C)	-15	-13	-6	2	8	14	18	17	11	6	-3	-11
			SOLAR (MJ/M**2/DAY)	5	9	14	18	20	23	23	19	13	8	5	4
83	TIMMINS	ONT	D-D (BASED ON 18 C)	1090	944	824	502	299	124	63	98	225	396	647	977
	(LATITUDE= 48.47)		MEAN AMB. TEMP. (C)	-17	-15	-9	1	9	15	17	16	11	5	-4	-13
			SOLAR (MJ/M**2/DAY)	5	9	14	17	19	22	21	18	12	7	4	3
84	TORONTO INT'L	ONT	D-D (BASED ON 18 C)	755	671	587	348	187	48	10	19	98	254	439	667
	(LATITUDE= 43.68)		MEAN AMB. TEMP. (C)	-6	-6	-1	6	12	18	21	20	16	10	3	-4
			SOLAR (MJ/M**2/DAY)	5	8	13	16	20	23	22	19	15	10	5	4

****************** CANADIAN CITY DATA USED IN F-CHART *** ****************

			JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
85	TRENTON ONT	D-D (BASED ON 18 C)	[illegible]	[illegible]	[illegible]	345	177	42	7	17	97	255	437	657
	(LATITUDE= 44.10)	MEAN AMB. TEMP. (C)	-7	-6	-1	7	12	16	21	20	16	10	3	-4
		SOLAR (MJ/M**2/DAY)	5	8	13	16	16	23	22	19	15	9	5	4
86	TRURO NS	D-D (BASED ON 18 C)	772	726	640	469	302	130	40	56	161	303	439	666
	(LATITUDE= 45.43)	MEAN AMB. TEMP. (C)	-6	-6	-2	3	9	13	18	17	13	8	3	-3
		SOLAR (MJ/M**2/DAY)	5	8	12	15	19	21	21	18	14	9	5	4
87	VAL D'OR QUE	D-D (BASED ON 18 C)	1079	931	827	506	3040	121	66	97	219	399	633	964
	(LATITUDE= 48.05)	MEAN AMB. TEMP. (C)	-17	-14	-8	1	8	15	17	15	11	5	-3	-13
		SOLAR (MJ/M**2/DAY)	5	9	15	17	19	22	21	17	12	8	4	4
88	VANCOUVER INT'L BC	D-D (BASED ON 18 C)	484	383	379	272	174	85	35	40	114	245	358	440
	(LATITUDE= 49.28)	MEAN AMB. TEMP. (C)	2	4	6	9	12	15	17	17	14	10	6	4
		SOLAR (MJ/M**2/DAY)	3	6	10	15	21	22	23	19	14	7	4	2
89	VICTORIA INT'L BC	D-D (BASED ON 18 C)	467	375	378	282	191	108	61	65	122	247	353	427
	(LATITUDE= 48.65)	MEAN AMB. TEMP. (C)	3	5	6	9	12	15	16	16	14	10	6	4
		SOLAR (MJ/M**2/DAY)	3	6	11	17	21	23	23	19	15	7	4	3
90	WHITEHORSE YUK	D-D (BASED ON 18 C)	1143	980	738	543	338	171	125	176	307	537	812	1048
	(LATITUDE= 60.75)	MEAN AMB. TEMP. (C)	-19	-13	-6	0	7	12	14	12	8	1	-9	-16
		SOLAR (MJ/M**2/DAY)	1	4	10	15	20	21	19	14	9	4	2	1
91	WIARTON ONT	D-D (BASED ON 18 C)	765	702	654	392	243	89	29	35	122	264	451	666
	(LATITUDE= 44.75)	MEAN AMB. TEMP. (C)	-7	-7	-3	5	10	16	19	18	15	10	3	-3
		SOLAR (MJ/M**2/DAY)	5	8	12	16	21	23	23	19	14	9	5	4
92	WILLIAMS LAKE BC	D-D (BASED ON 18 C)	868	592	567	416	283	138	89	102	228	393	611	798
	(LATITUDE= 52.13)	MEAN AMB. TEMP. (C)	-10	-5	-4	4	10	13	16	15	11	5	-2	-7
		SOLAR (MJ/M**2/DAY)	3	6	11	16	20	22	22	18	13	7	4	2
93	WINDSOR ONT	D-D (BASED ON 18 C)	691	606	520	295	141	29	3	8	64	204	411	618
	(LATITUDE= 42.33)	MEAN AMB. TEMP. (C)	-4	-3	1	8	14	20	22	21	17	12	4	-2
		SOLAR (MJ/M**2/DAY)	5	8	12	16	20	23	23	19	16	11	5	5
94	WINNIPEG INT'L MAN	D-D (BASED ON 18 C)	1126	953	810	44	226	76	21	38	175	356	674	993
	(LATITUDE= 49.90)	MEAN AMB. TEMP. (C)	-18	-16	-8	3	11	17	20	19	13	7	-4	-14
		SOLAR (MJ/M**2/DAY)	5	9	14	17	21	22	23	19	13	8	5	4
95	YARMOUTH NS	D-D (BASED ON 18 C)	641	592	551	401	270	141	55	126	252	252	376	564
	(LATITUDE= 43.83)	MEAN AMB. TEMP. (C)	-3	-3	0	5	9	13	16	16	14	10	5	0
		SOLAR (MJ/M**2/DAY)	5	9	13	15	19	21	21	19	15	9	5	5
96	YELLOWKNIFE NWT	D-D (BASED ON 18 C)	1445	1235	1138	777	435	178	76	126	333	596	967	1287
	(LATITUDE= 62.47)	MEAN AMB. TEMP. (C)	-29	-26	-19	-8	4	12	16	14	7	-1	-14	-24
		SOLAR (MJ/M**2/DAY)	1	4	10	18	24	23	20	16	8	4	1	1
97	YORKTON SASK	D-D (BASED ON 18 C)	1163	952	853	480	256	102	34	58	211	400	722	1008
	(LATITUDE= 51.18)	MEAN AMB. TEMP. (C)	-19	-16	-10	2	10	15	19	17	11	5	-6	-14
		SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	24	19	13	9	5	3

APPENDIX 2A-2

$\overline{R}$ AS A FUNCTION OF $\overline{K}_T$

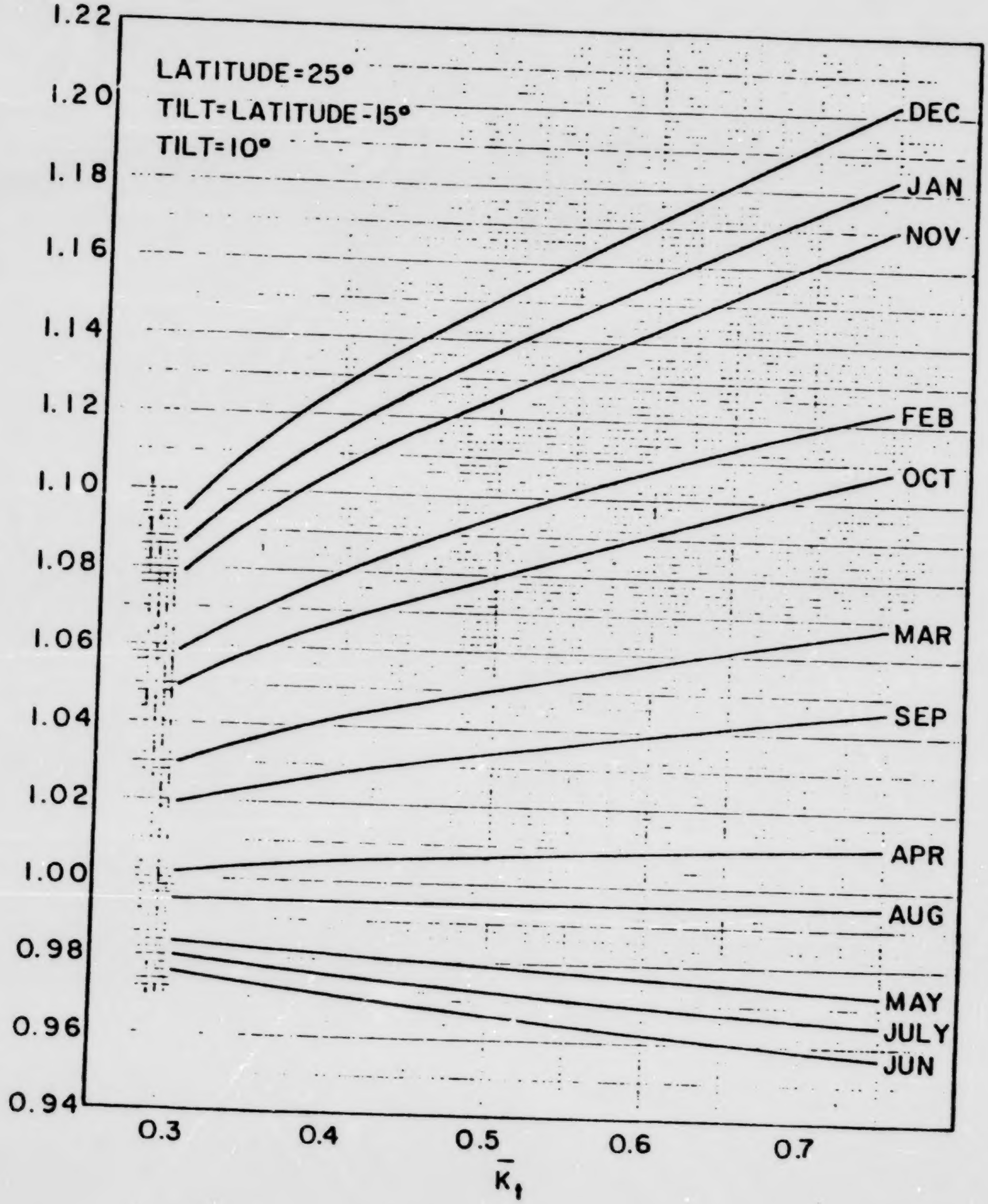

Figure 2A-2-1. Tilt Correction Factor for 25° Latitude, 10° Tilt

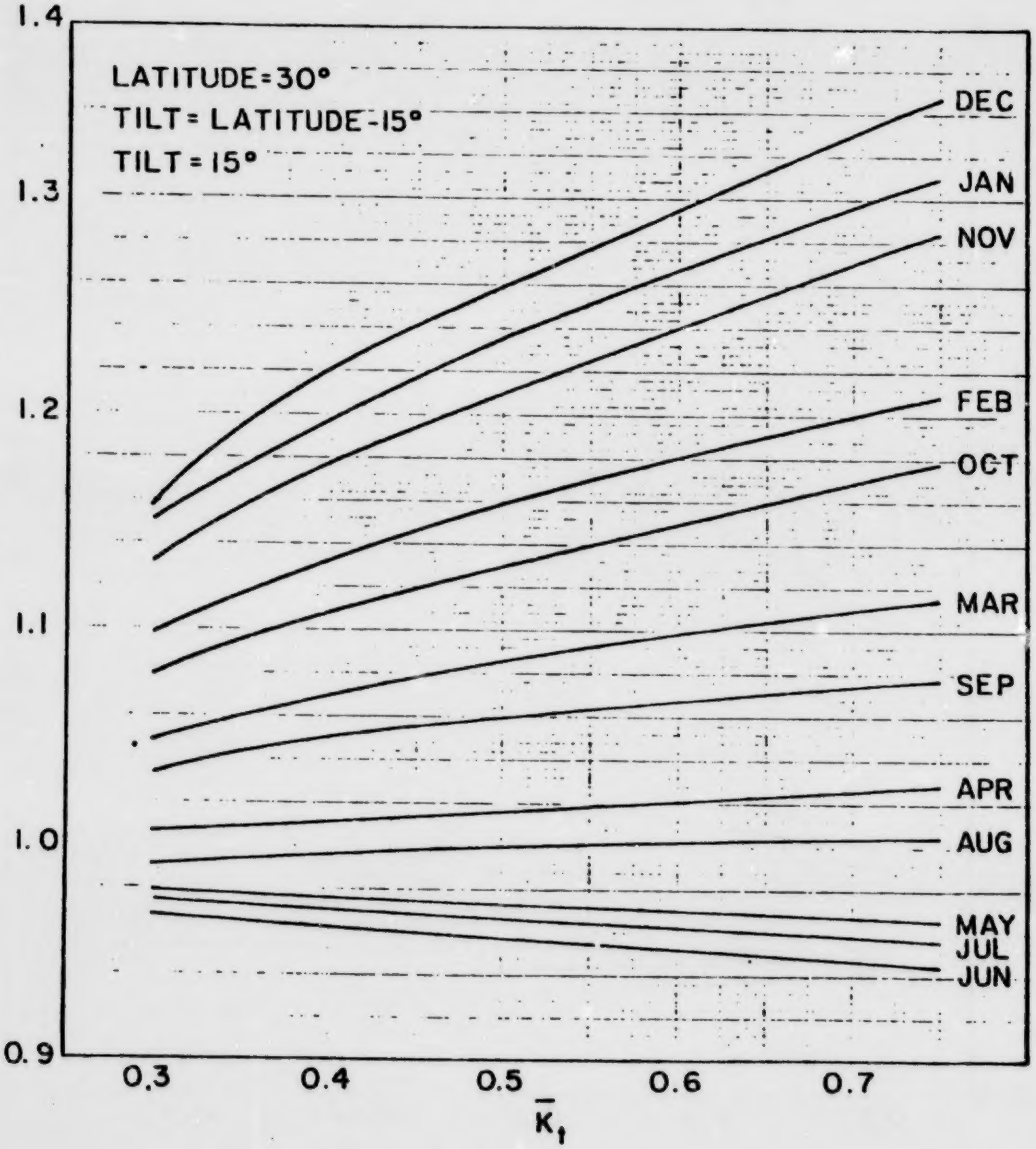

Figure 2A-2-2. Tilt Correction Factor for 30° Latitude, 15° Tilt

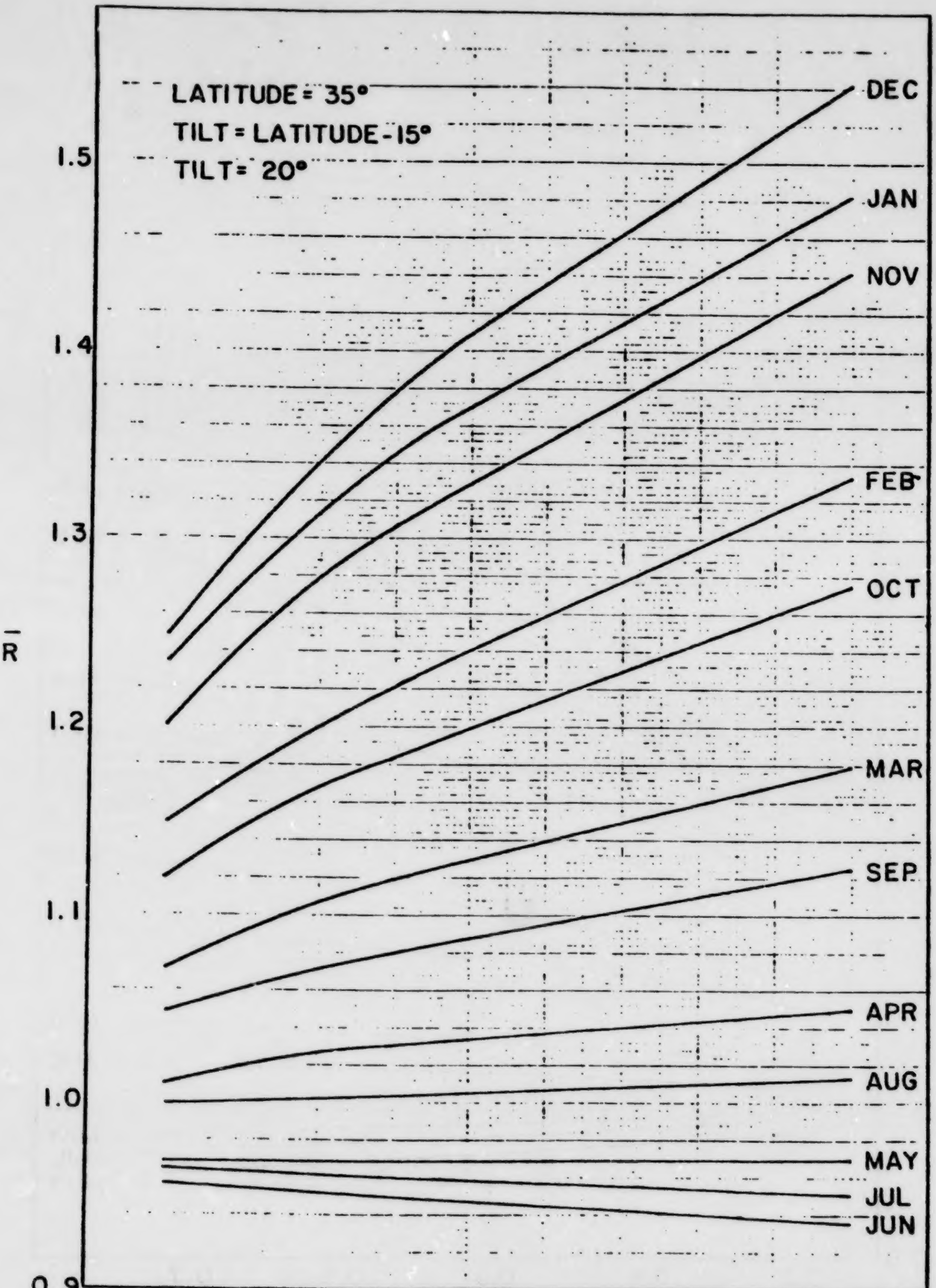

Figure 2A-2-3. Tilt Correction Factor for 35° Latitude, 20° Tilt

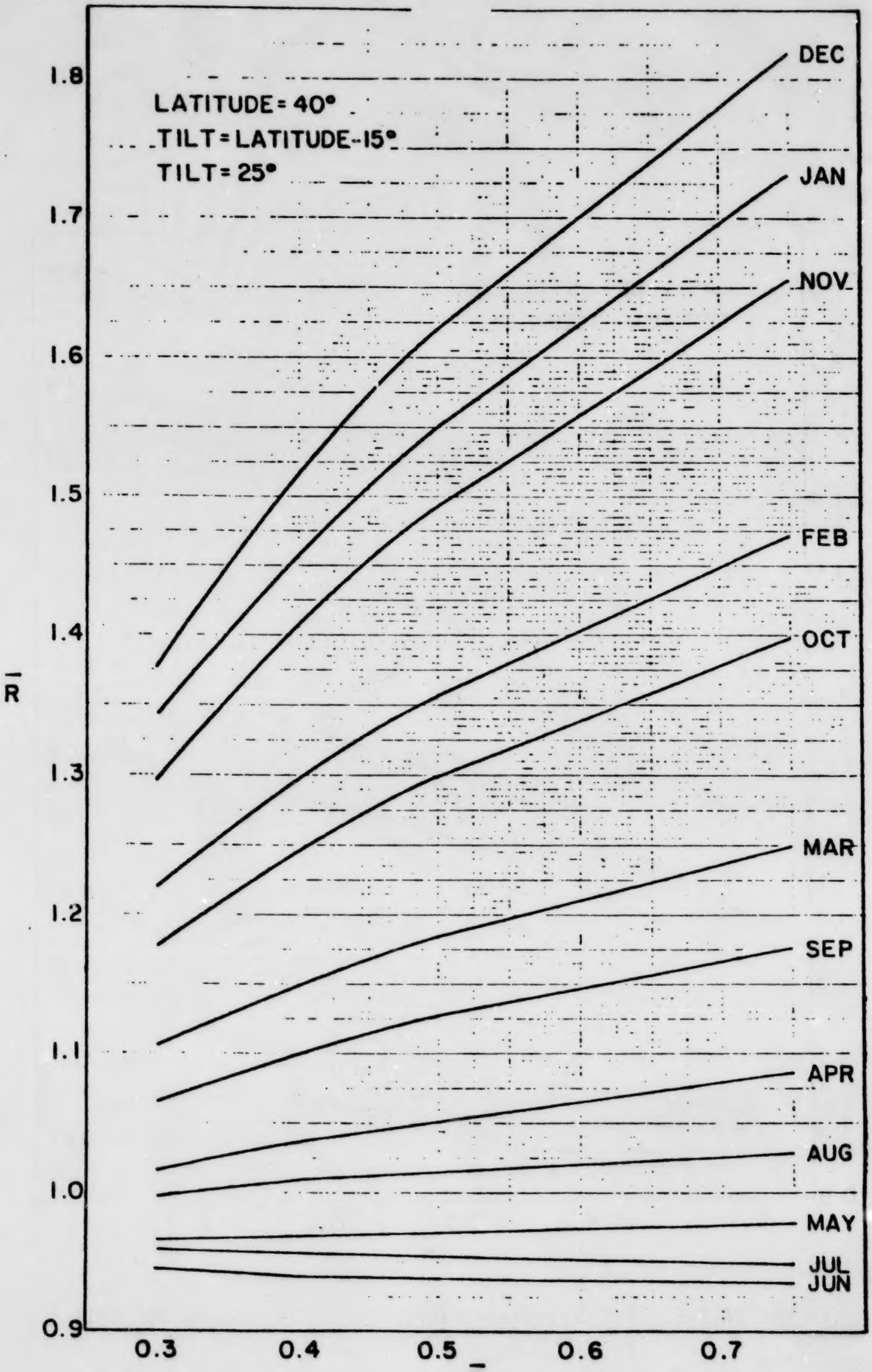

Figure 2A-2-4. Tilt Correction Factor for 40° Latitude, 25° Tilt

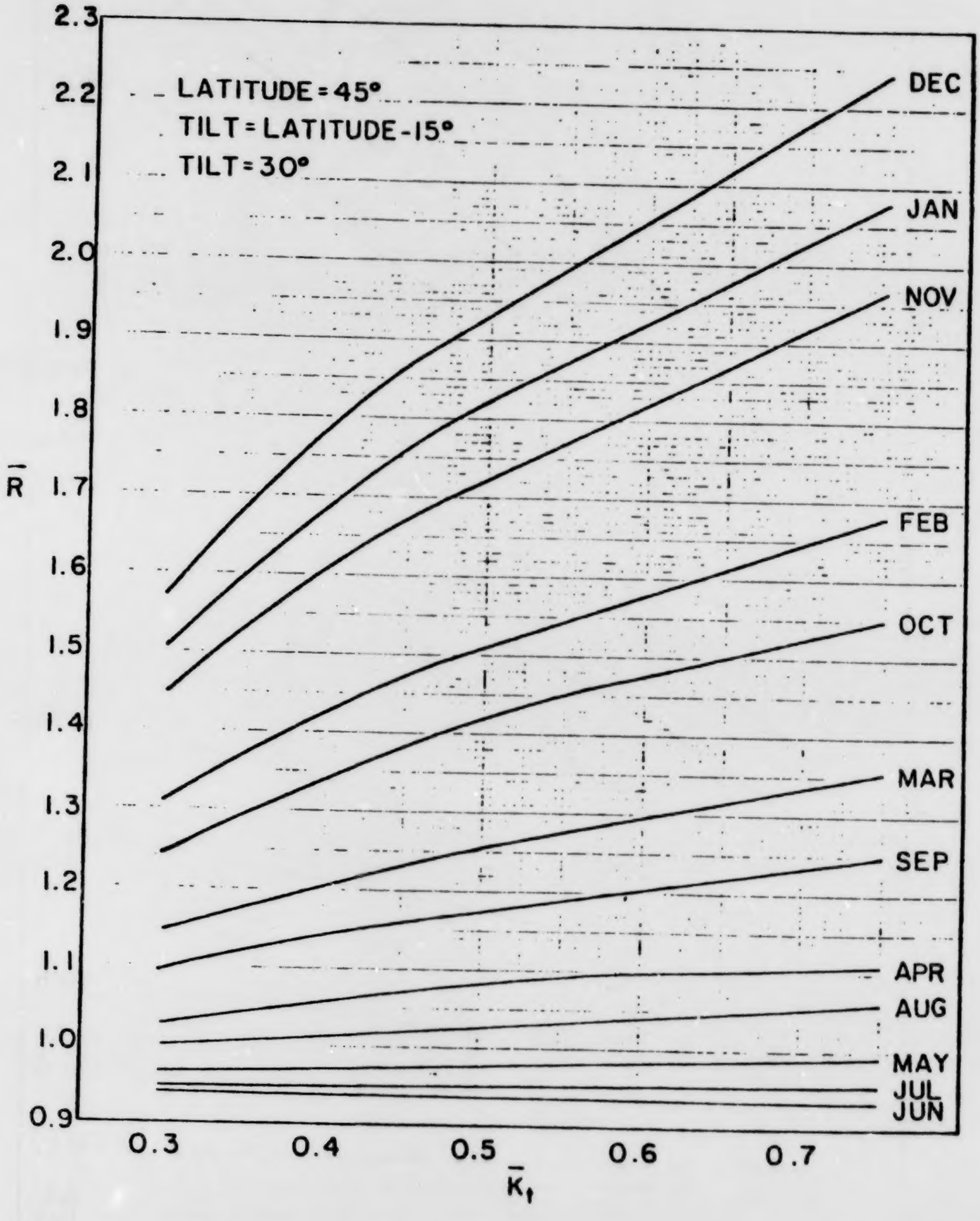

Figure 2A-2-5. Tilt Correction Factor for 45° Latitude, 30° Tilt

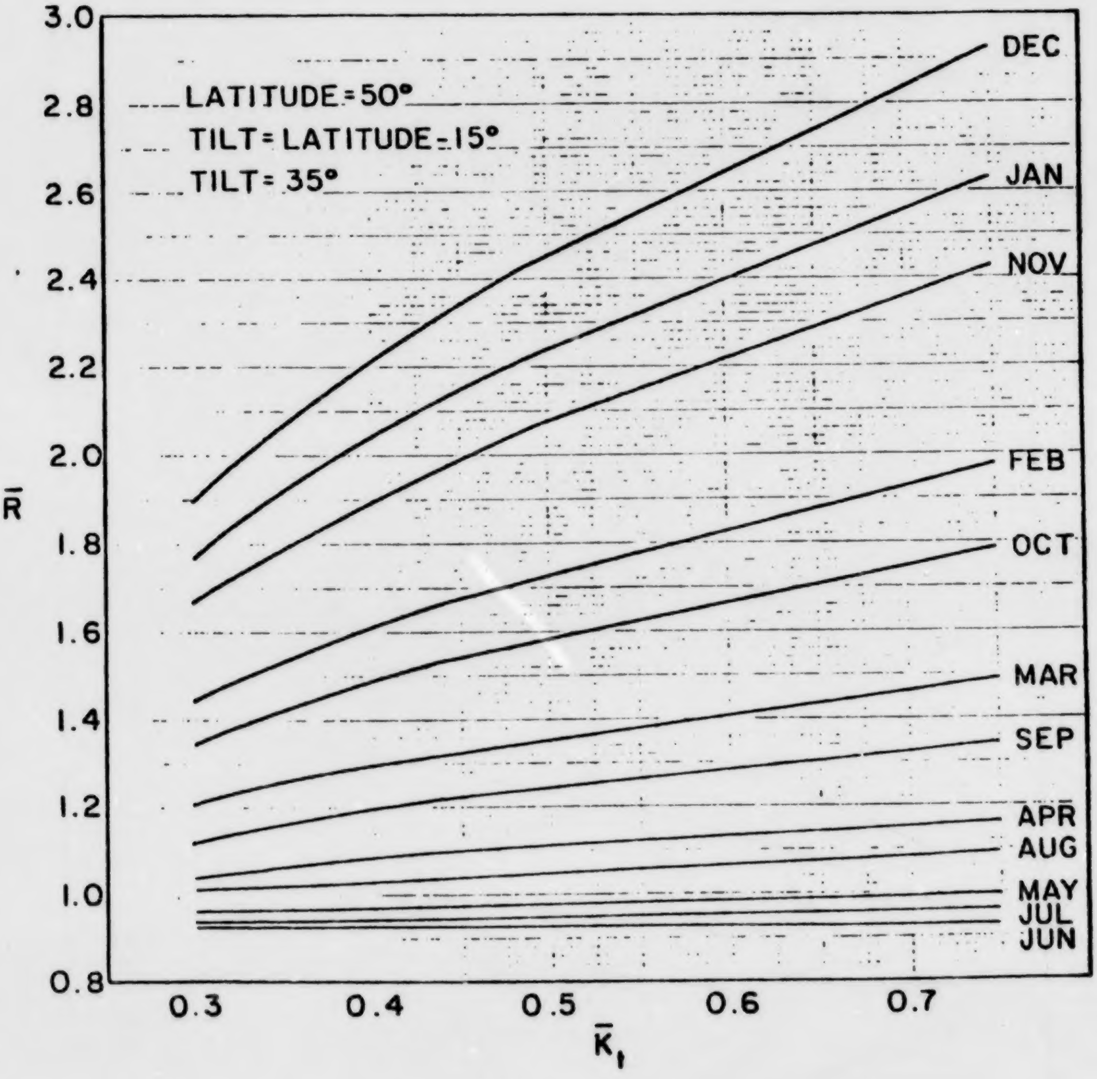

Figure 2A-2-6. Tilt Correction Factor for 50° Latitude, 35° Tilt

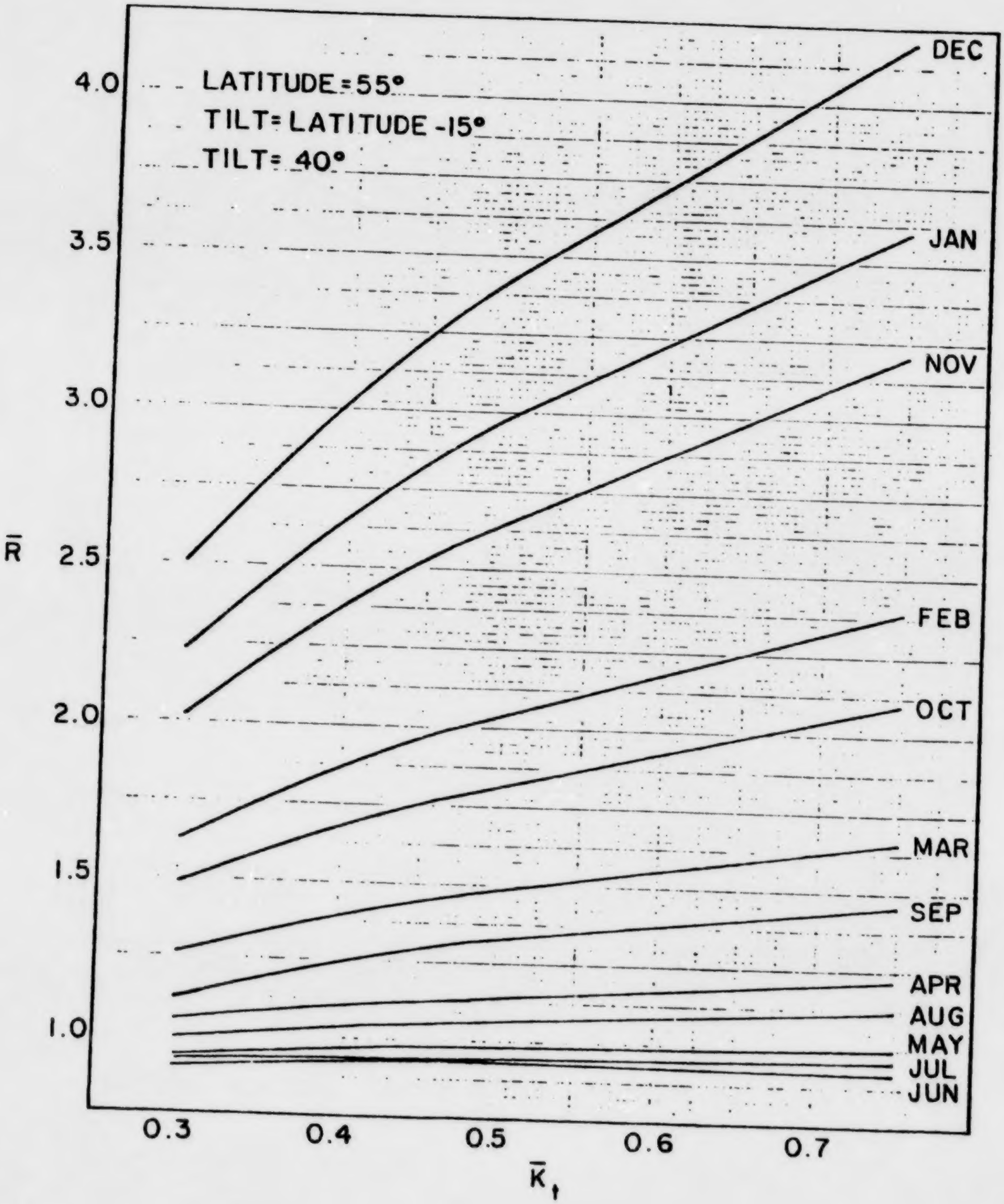

Figure 2A-2-7. Tilt Correction Factor for 55° Latitude, 40° Tilt

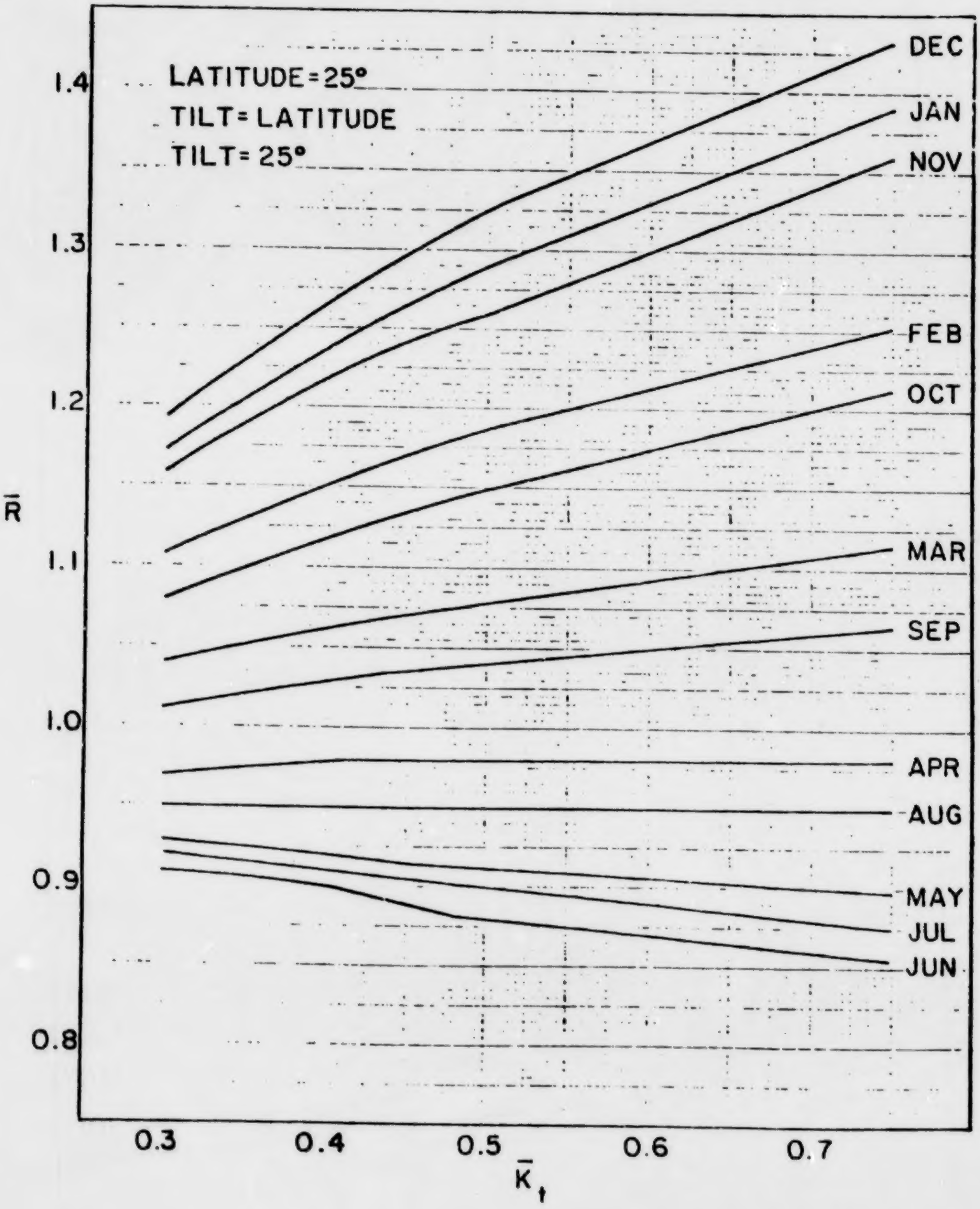

Figure 2A-2-8. Tilt Correction Factor for 25° Latitude, 25° Tilt

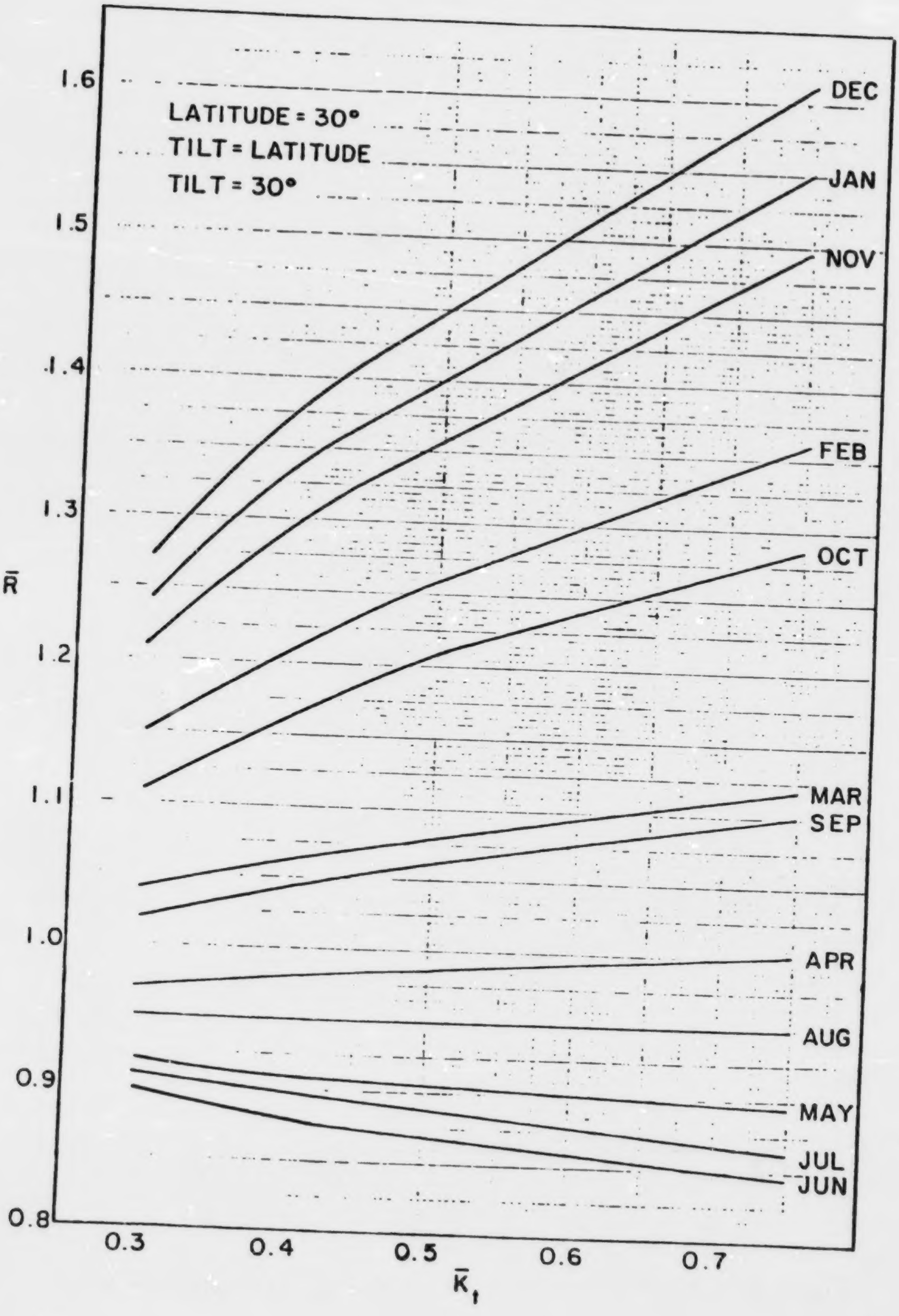

Figure 2A-2-9. Tilt Correction Factor for 30° Latitude, 30° Tilt

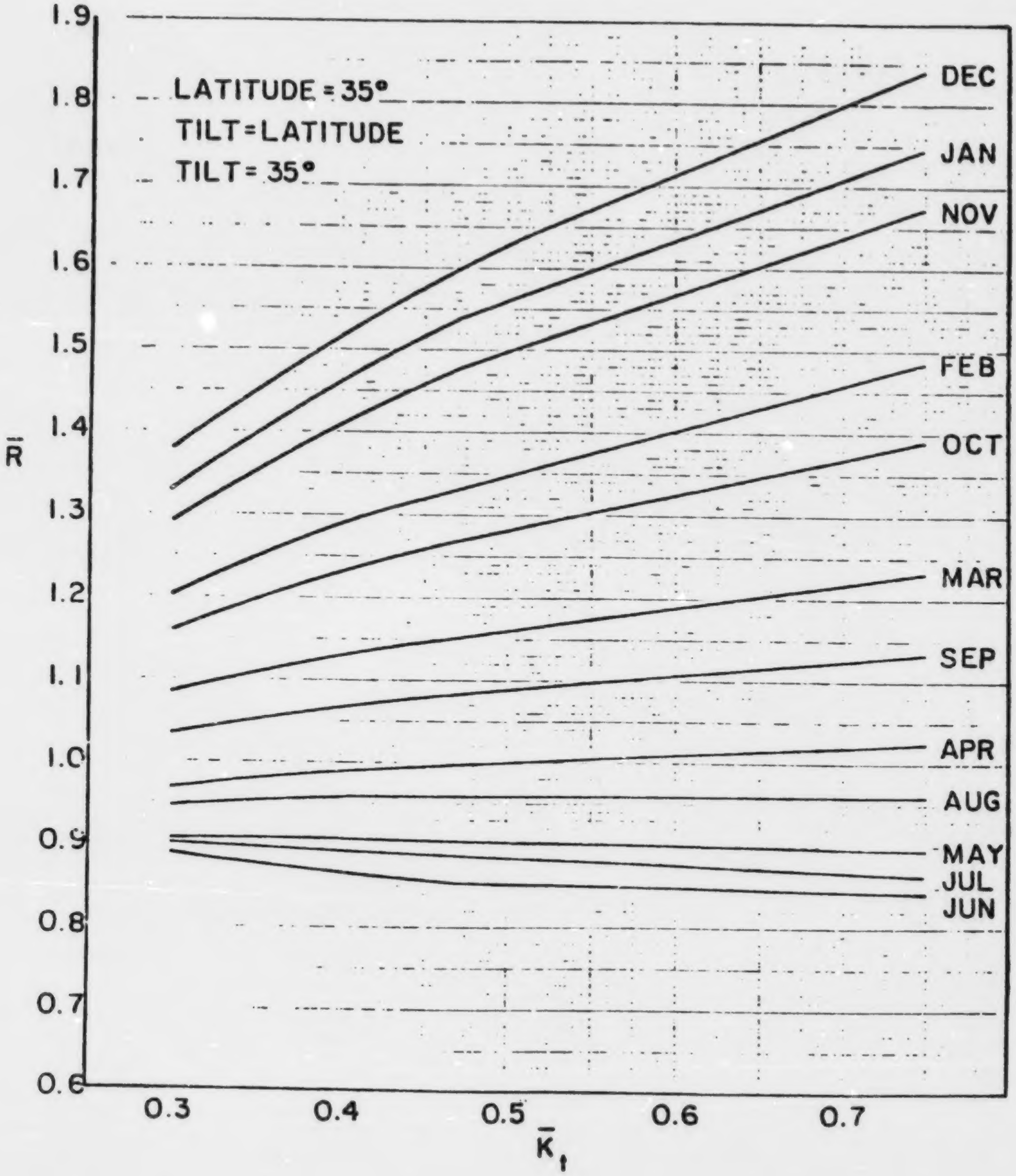

Figure 2A-2-10. Tilt Correction Factor for 35° Latitude, 35° Tilt

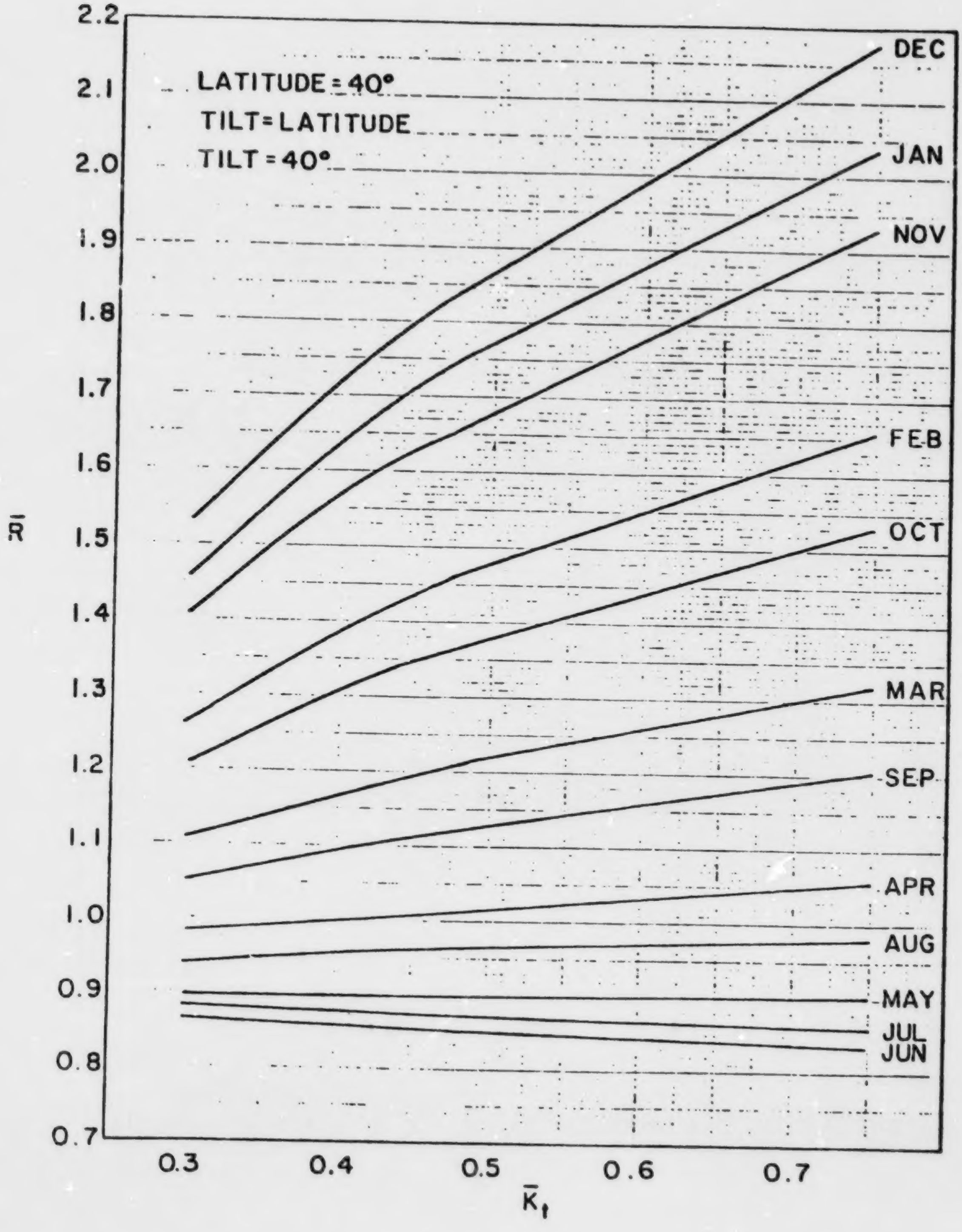

Figure 2A-2-11. Tilt Correction Factor for 40° Latitude, 40° Tilt

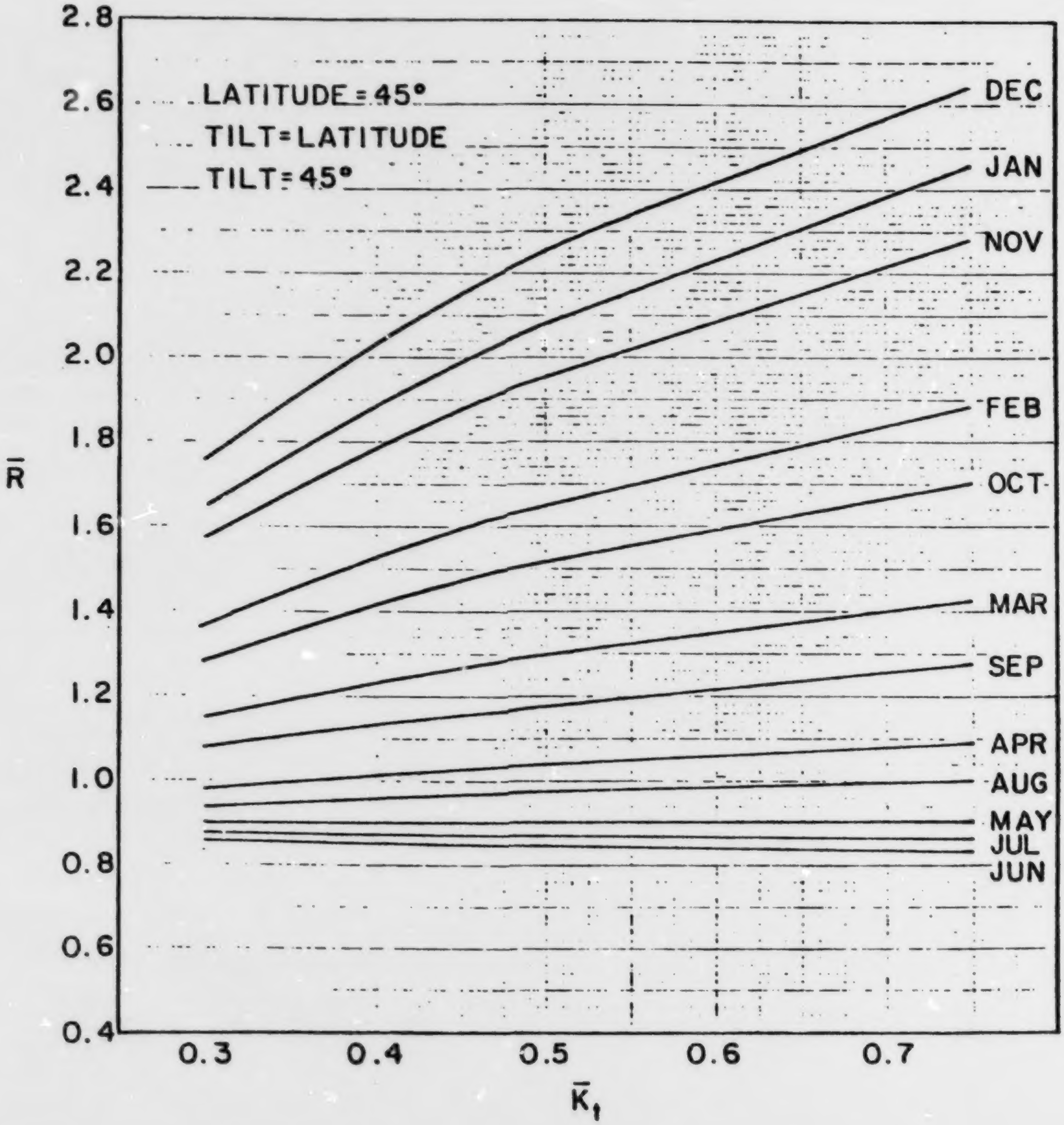

Figure 2A-2-12. Tilt Correction Factor for 45° Latitude, 45° Tilt

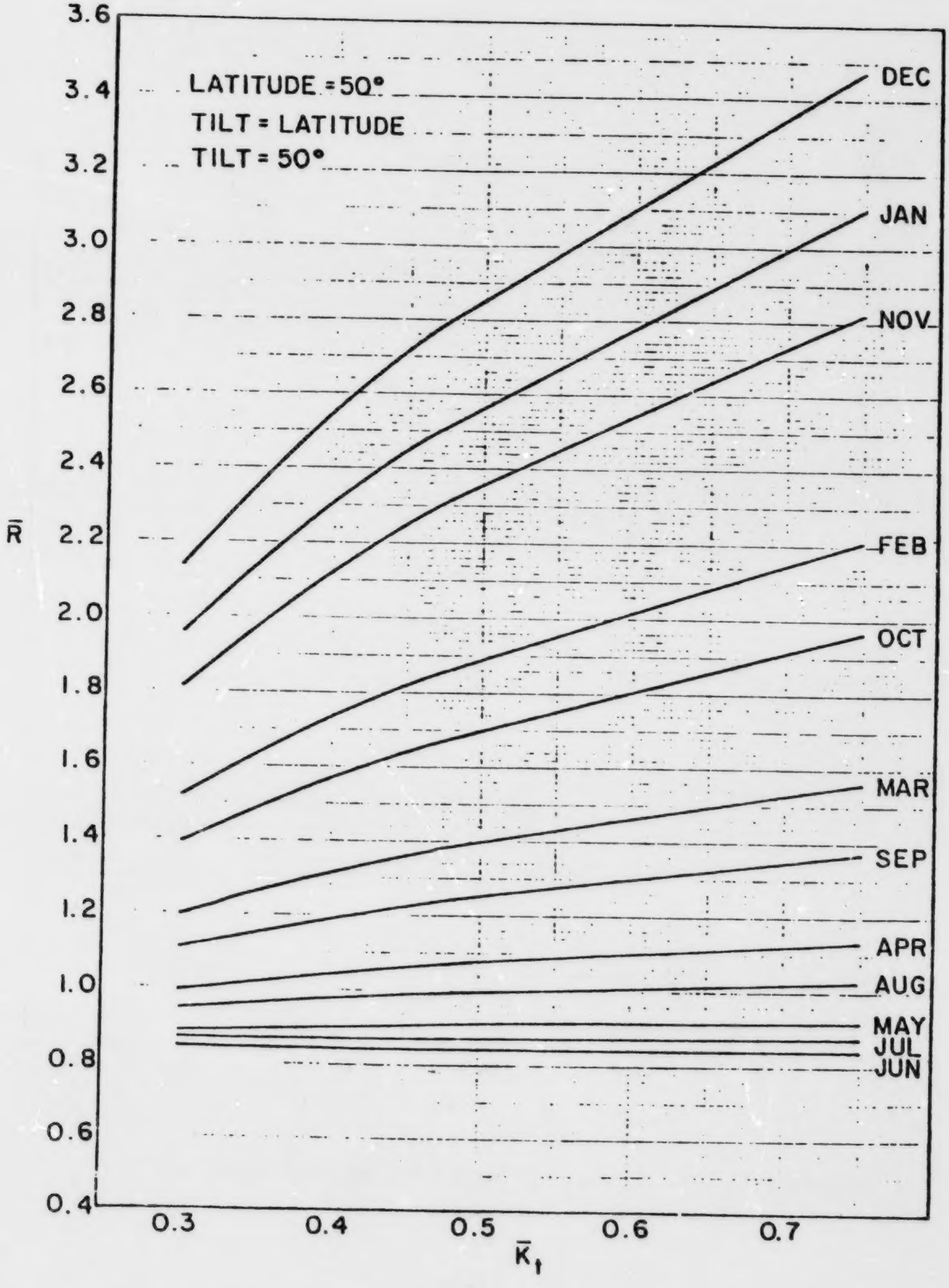

Figure 2A-2-13. Tilt Correction Factor for 50° Latitude, 50° Tilt

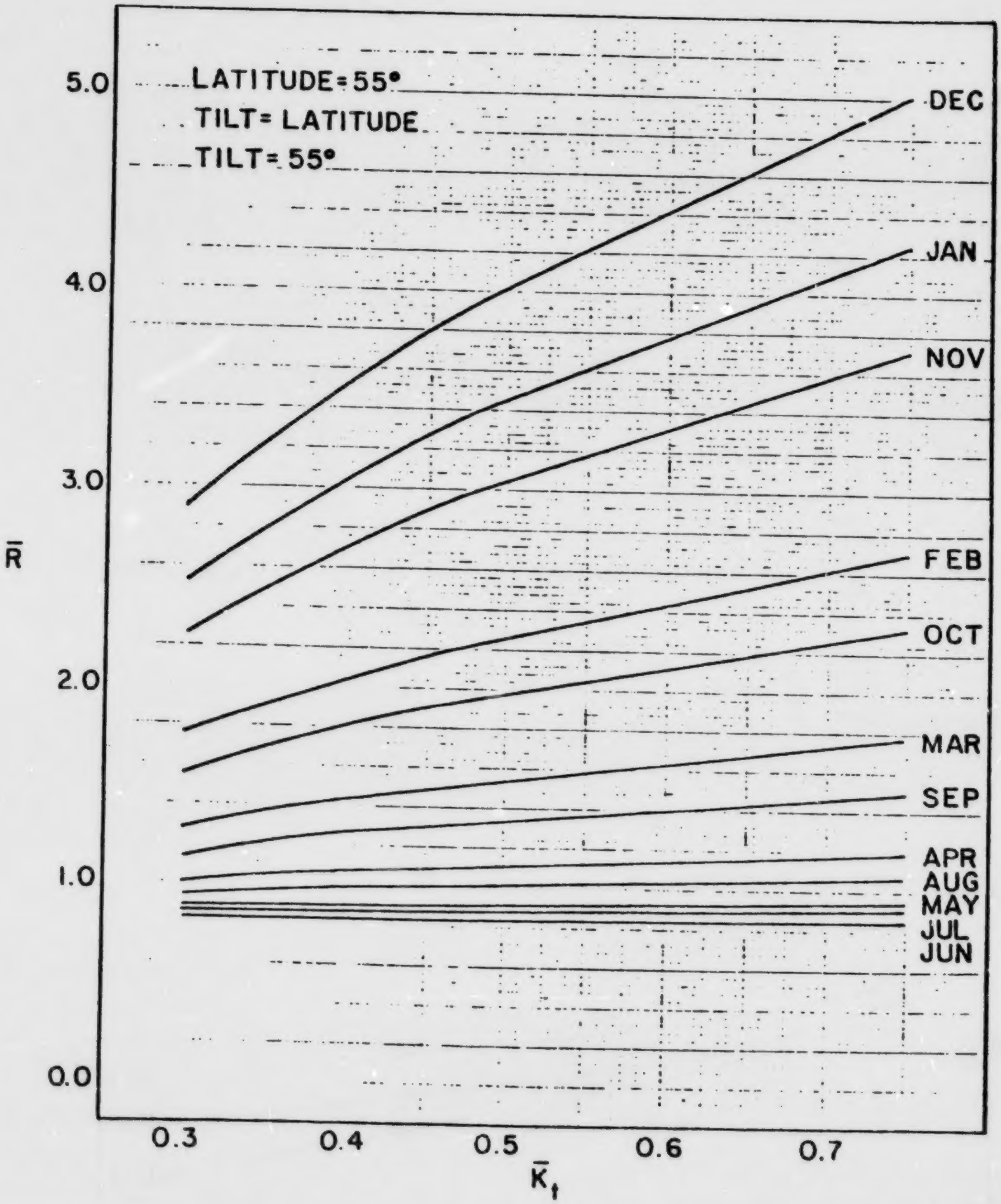

Figure 2A-2-14. Tilt Correction Factor for 55° Latitude, 55° Tilt

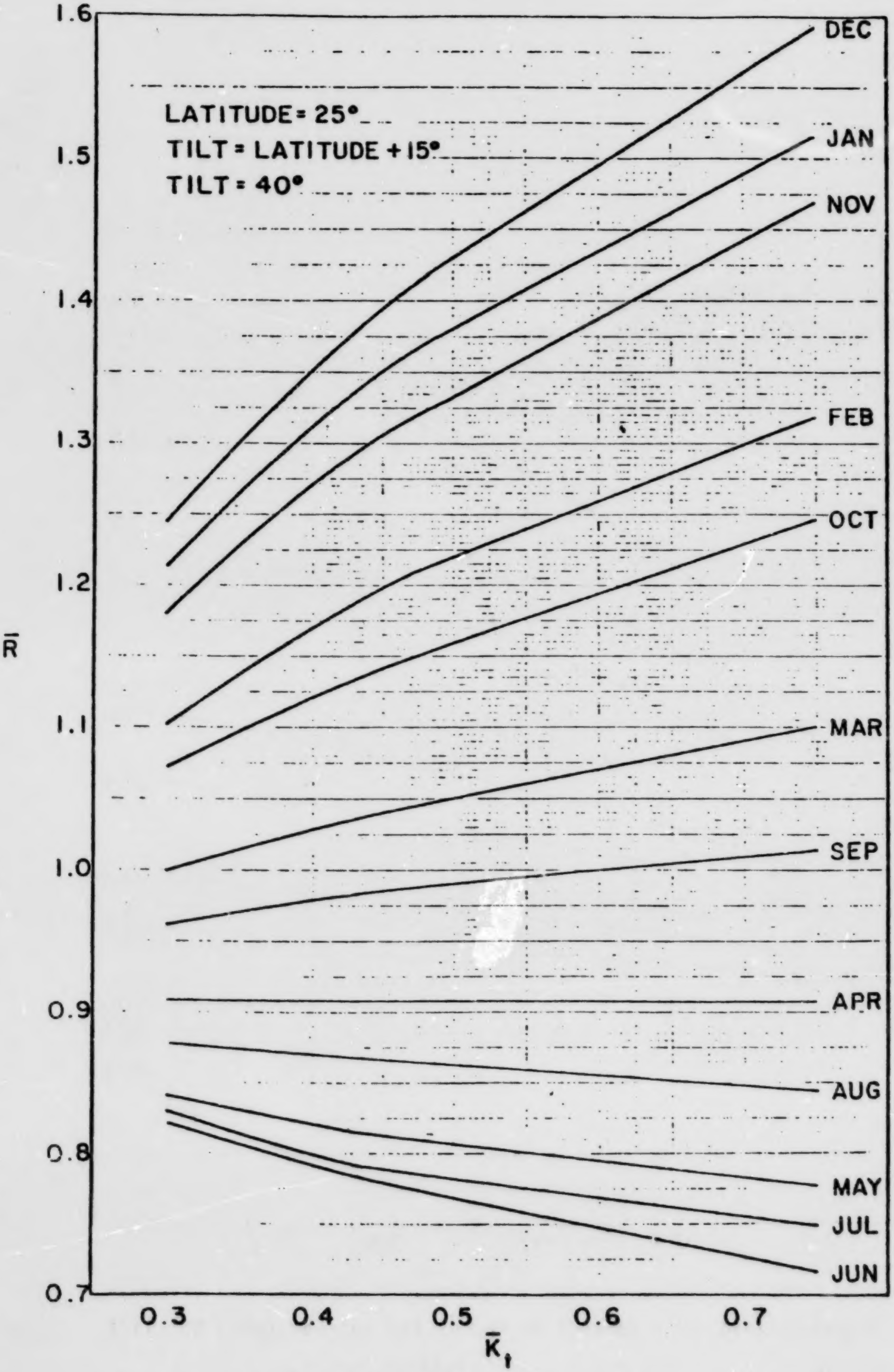

Figure 2A-2-15. Tilt Correction Factor for 25° Latitude, 40° Tilt

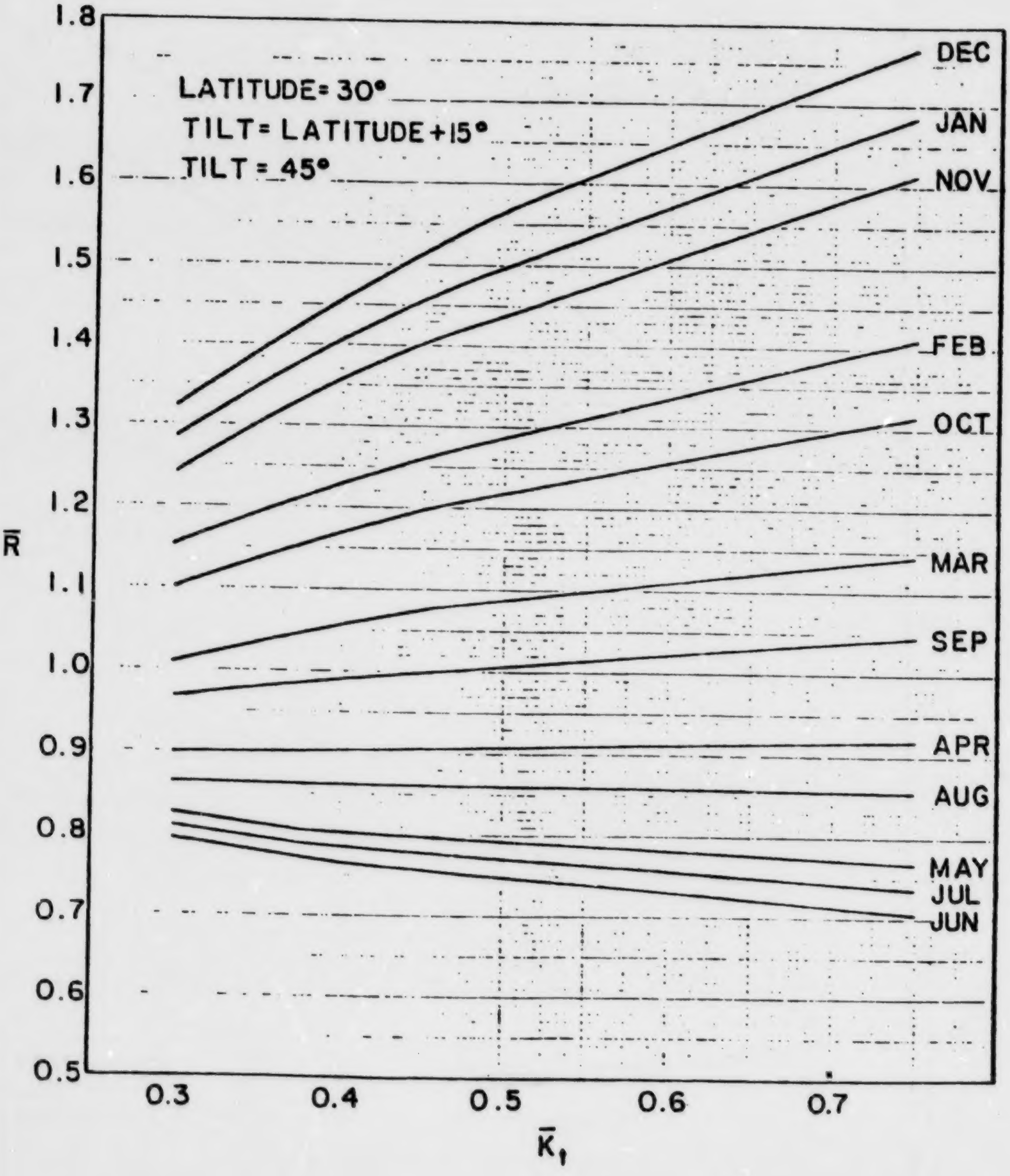

Figure 2A-2-16. Tilt Correction Factor for 30° Latitude, 45° Tilt

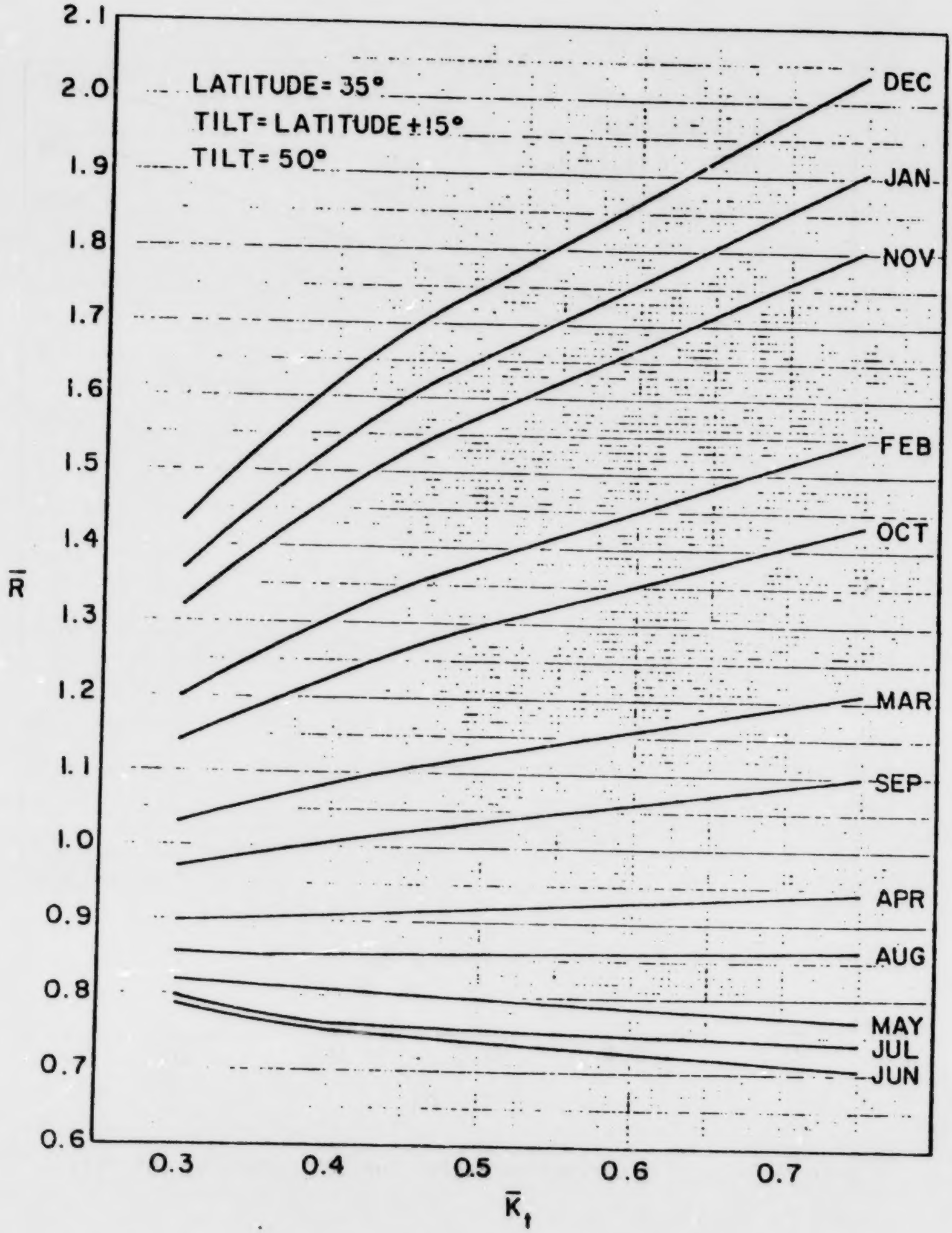

Figure 2A-2-17. Tilt Correction Factor for 35° Latitude, 50° Tilt

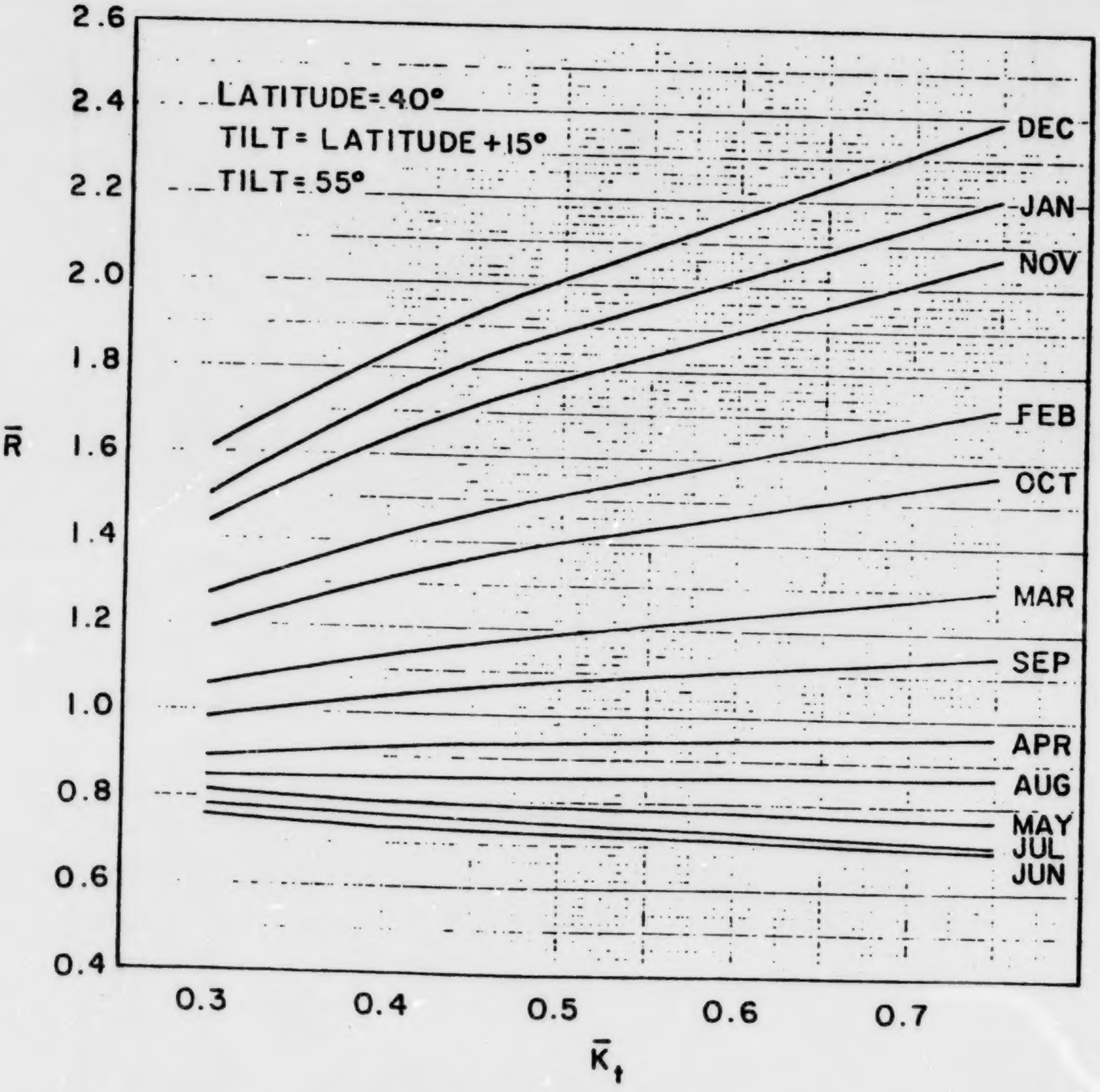

Figure 2A-2-18. Tilt Correction Factor for 40° Latitude, 55° Tilt

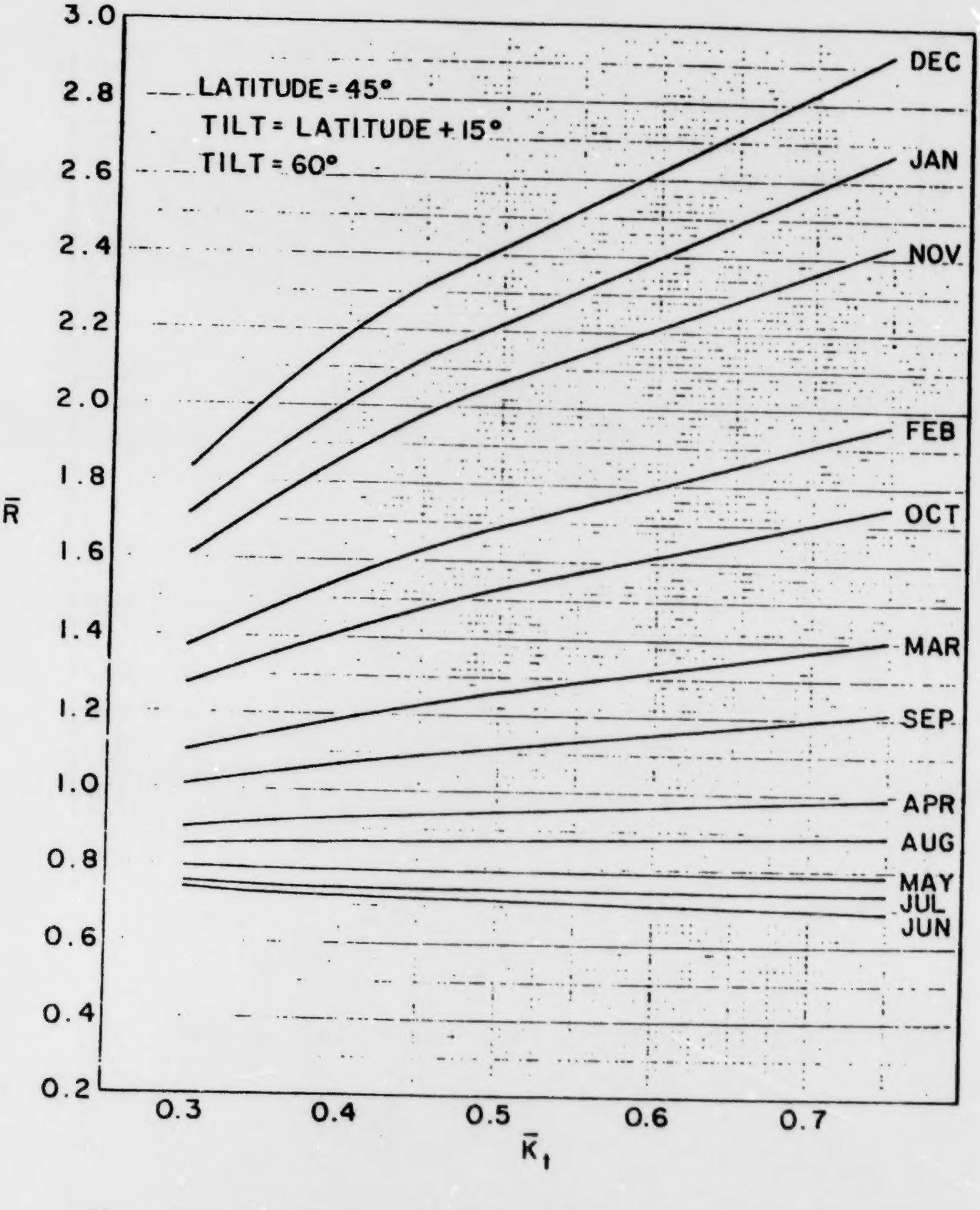

Figure 2A-2-19. Tilt Correction Factor for 45° Latitude, 60° Tilt

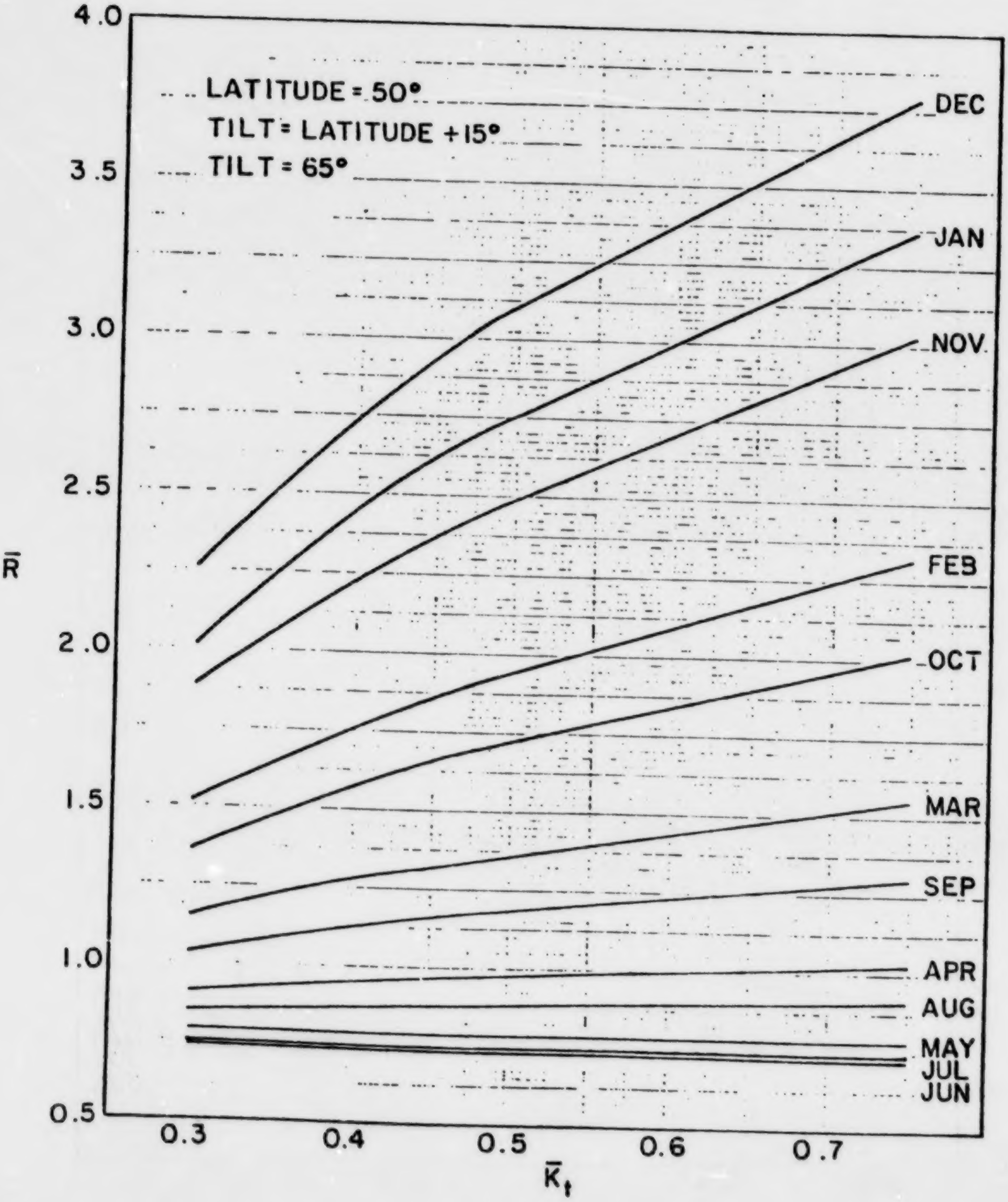

Figure 2A-2-20. Tilt Correction Factor for 50° Latitude, 65° Tilt

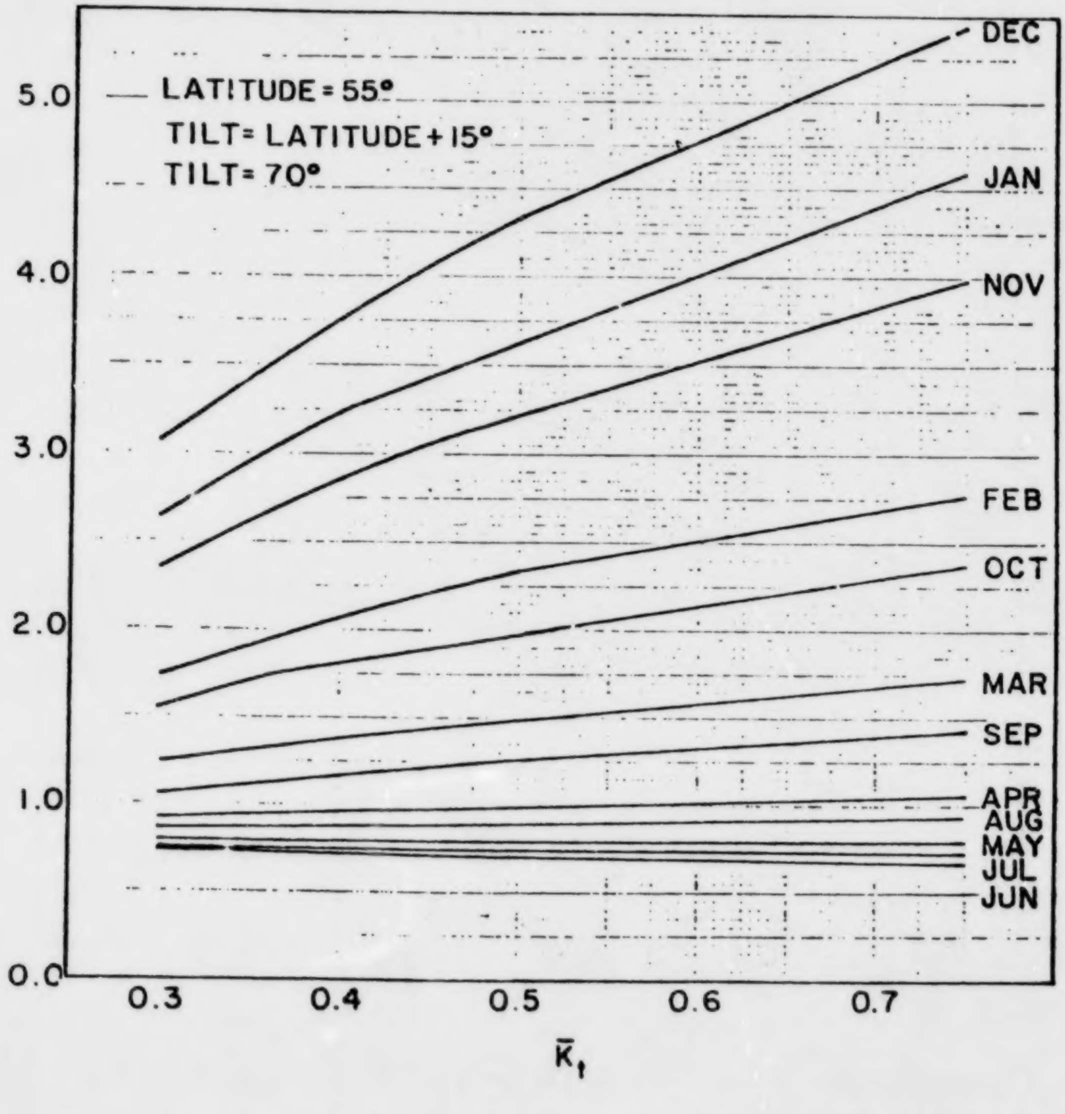

Figure 2A-2-21. Tilt Correction Factor for 55° Latitude, 70° Tilt

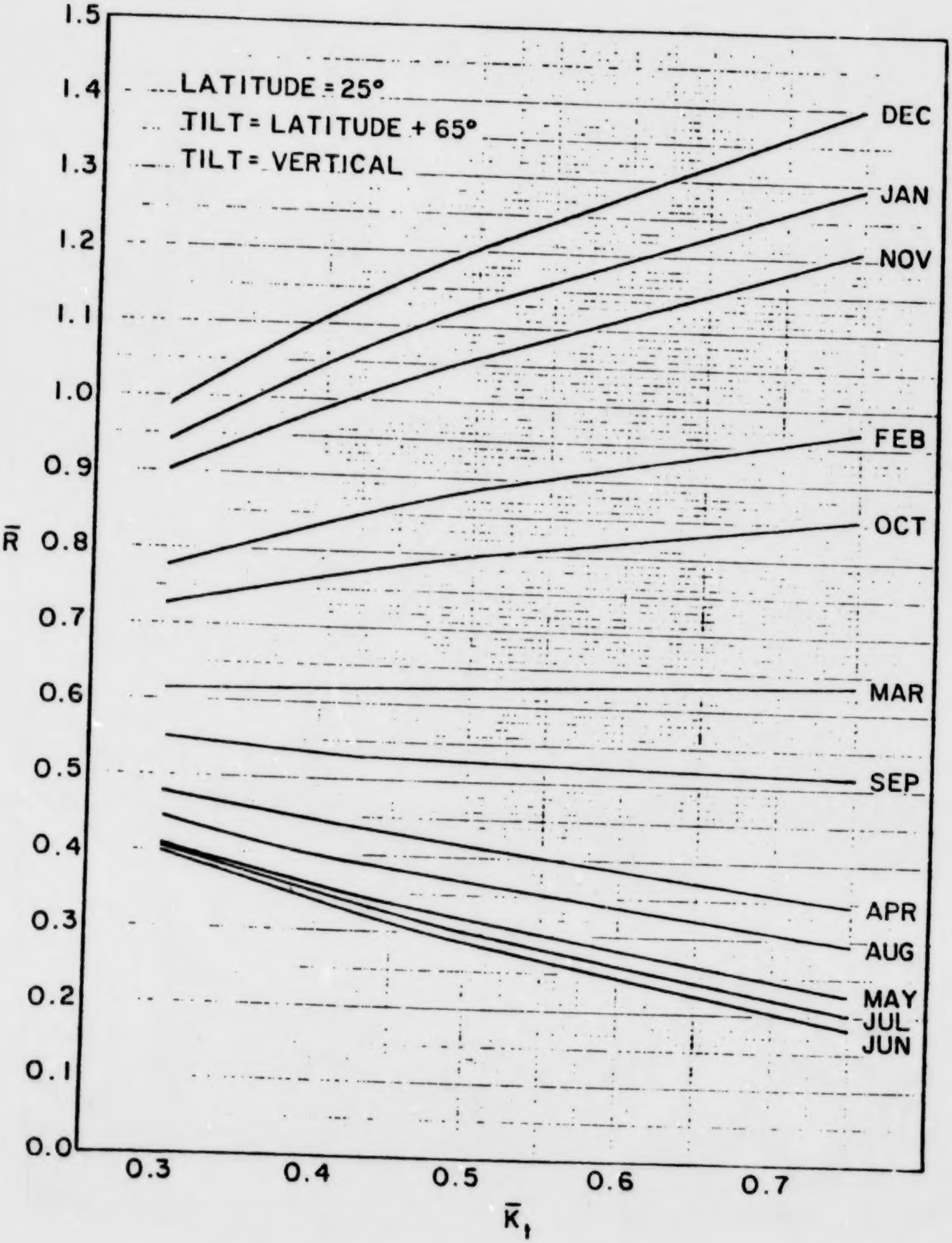

Figure 2A-2-22.Tilt Correction Factor for 25° Latitude, Vertical Tilt

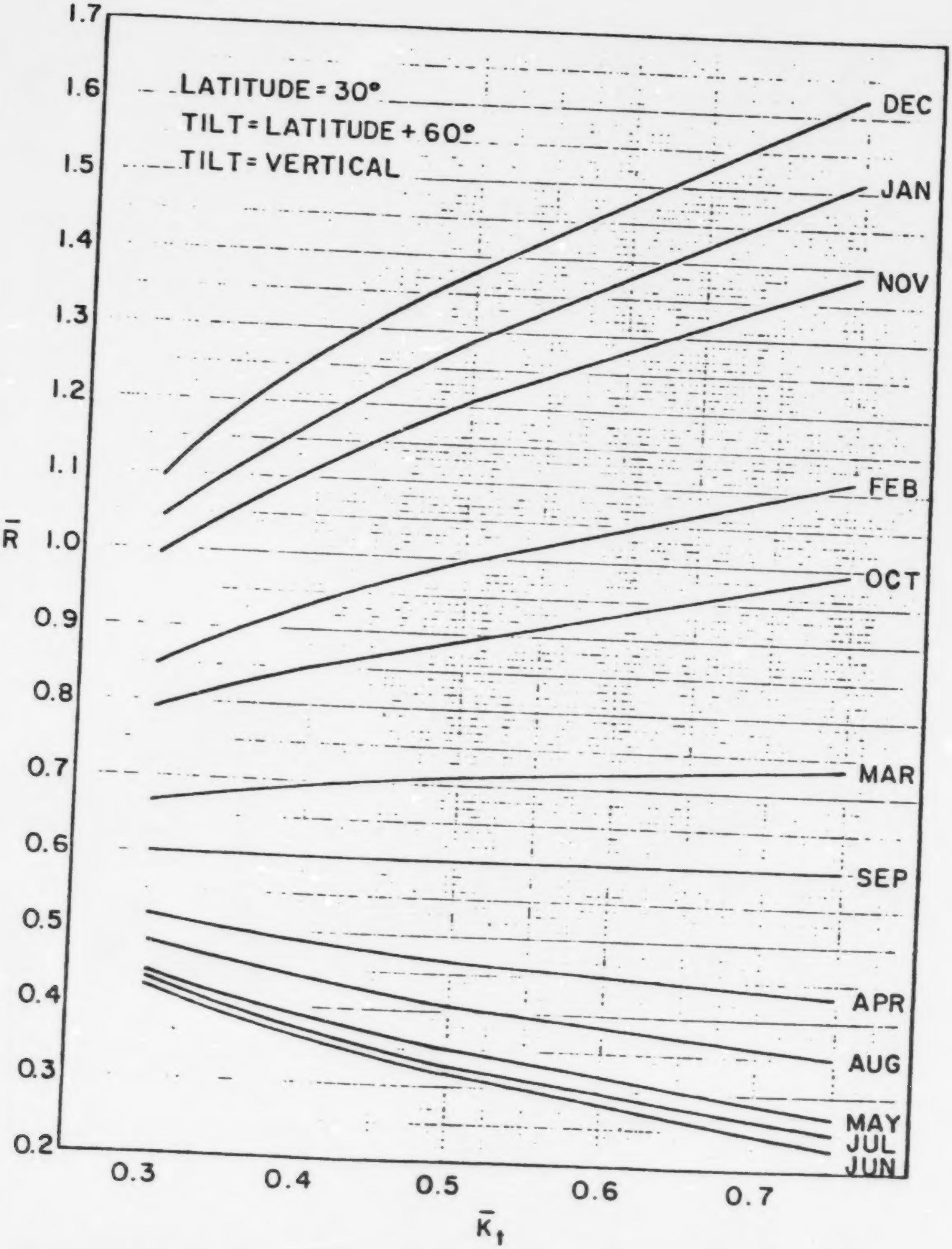

Figure 2A-2-23. Tilt Correction Factor for 30° Latitude, Vertical Tilt

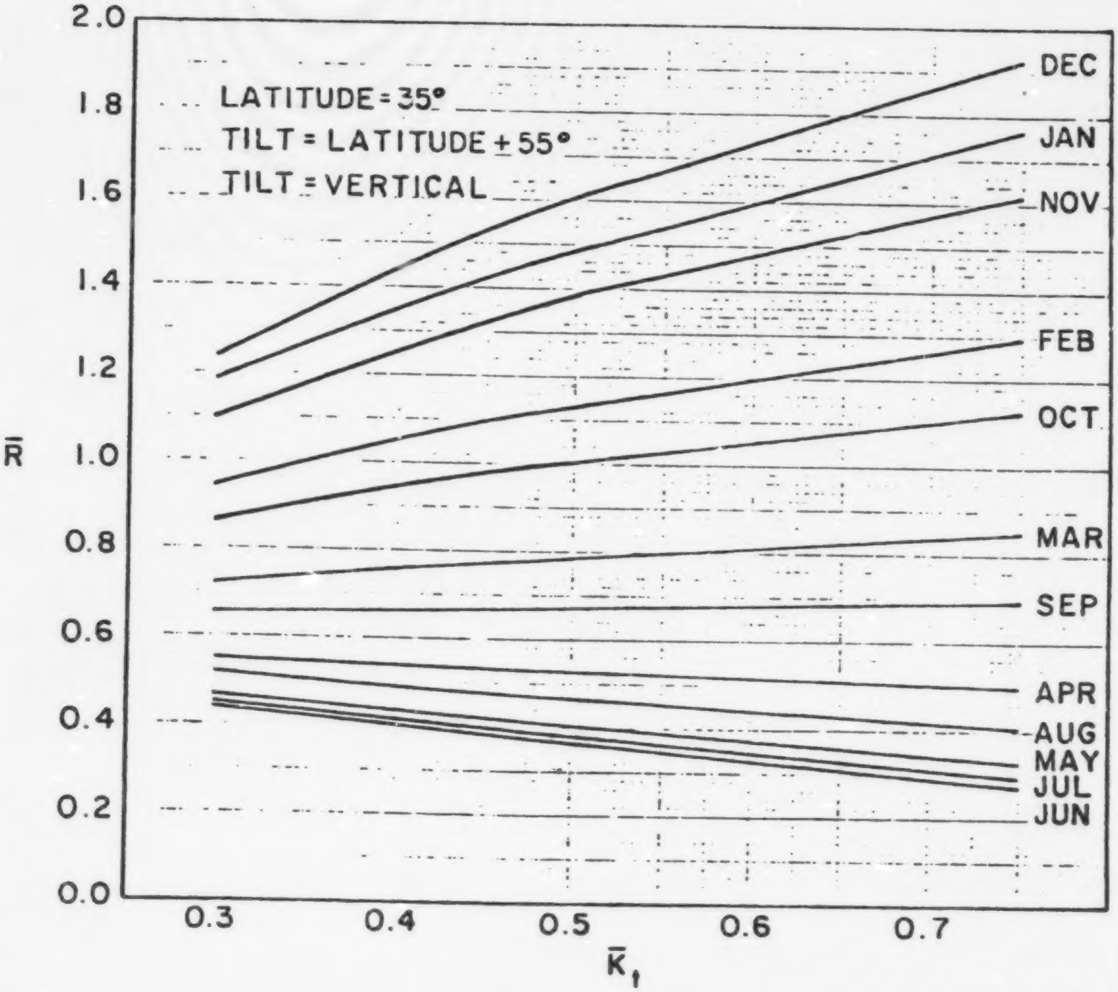

Figure 2A-2-24. Tilt Correction Factor for 35° Latitude, Vertical Tilt

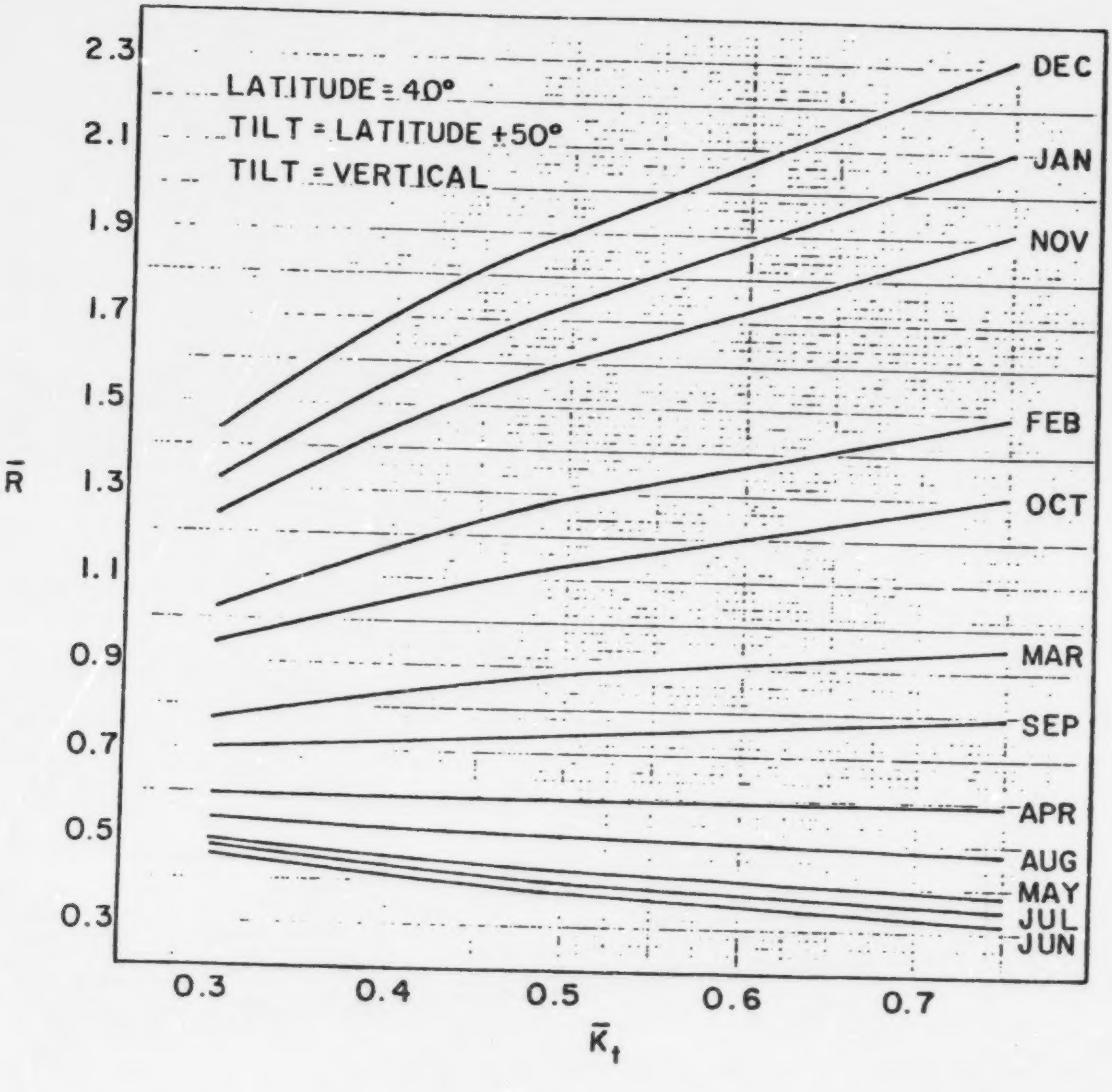

Figure 2A-2-25. Tilt Correction Factor for 40° Latitude, Vertical Tilt

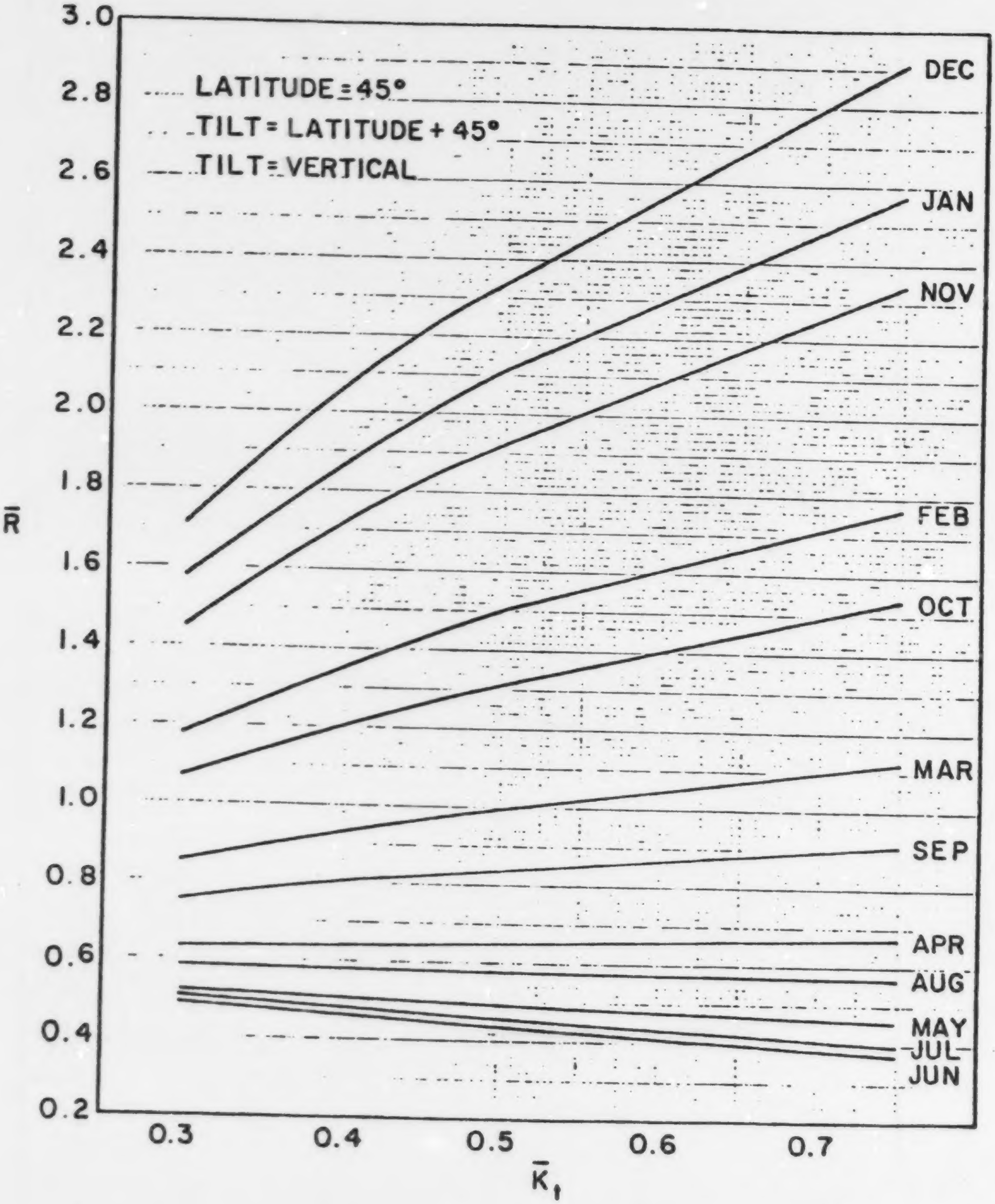

Figure 2A-2-26. Tilt Correction Factor for 45° Latitude, Vertical Tilt

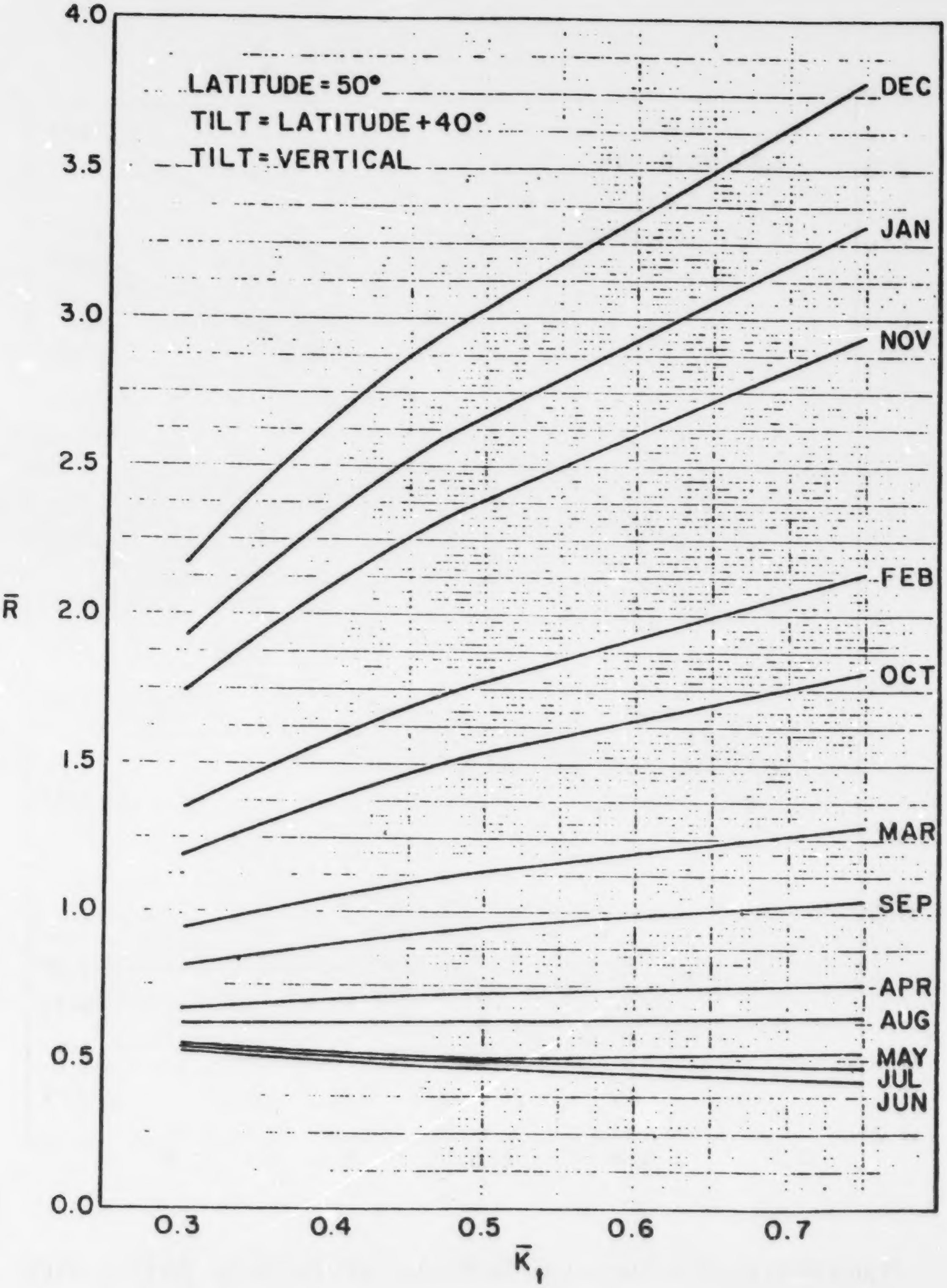

Figure 2A-2-27. Tilt Correction Factor for 50° Latitude, Vertical Tilt

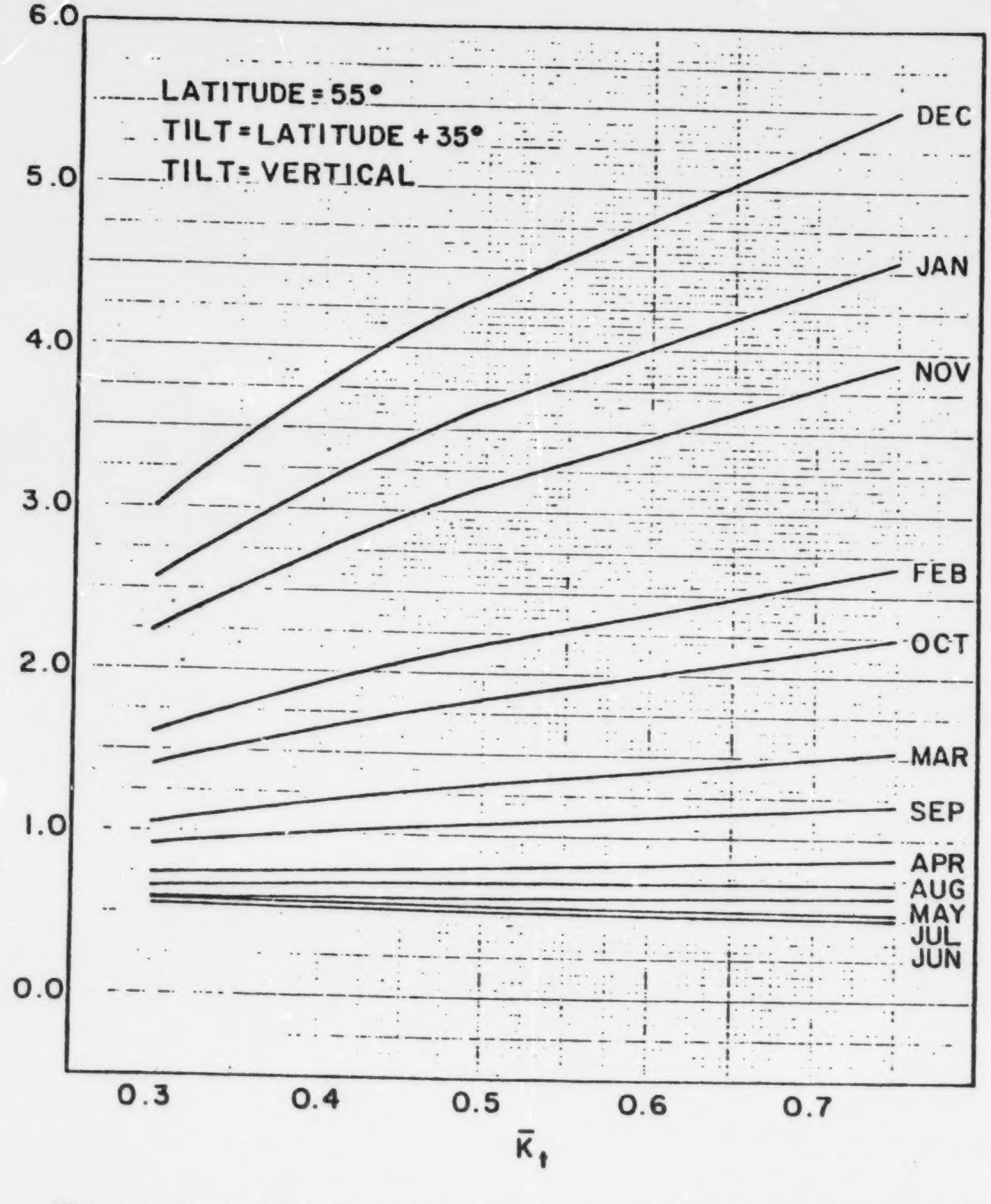

Figure 2A-2-28. Tilt Correction Factor for 55° Latitude, Vertical Tilt

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CHAPTER 3

HEAT TRANSFER AND THE EQUATIONS OF SOLAR COLLECTOR PERFORMANCE

3.1 INTRODUCTION

The topics covered in this chapter are:

1. The transmission of radiation through collector covers, and absorbtion by absorber surfaces.
2. Heat transfer in flat plate collectors.
3. The collector efficiency equation.

The material presented in this chapter is covered in more detail in Chapters 5, 6 and 7 of D and B. The objective of these notes is to provide the student with a general understanding of the thermal processes occurring in a flat plate collector, not to develop his capability to carry out quantitative analysis of these processes. The analysis presented is for a liquid cooled tube in plate collector of conventional design, but the same approach can be applied to other collector types.

3.2 TRANSMISSION AND ABSORBTION IN FLAT PLATE COLLECTORS

3.2.1 Transmission Through Partially Transparent Media

Figure 3.2.1 shows a beam of radiation passing from a medium with refractive index n_1 to one with refractive index n_2

Snell's law

$$\frac{n_1}{n_2} = \frac{\sin\theta_2}{\sin\theta_1} \qquad \text{(Eqn 3.2.1)}$$

relates the angles to the refractive indices.

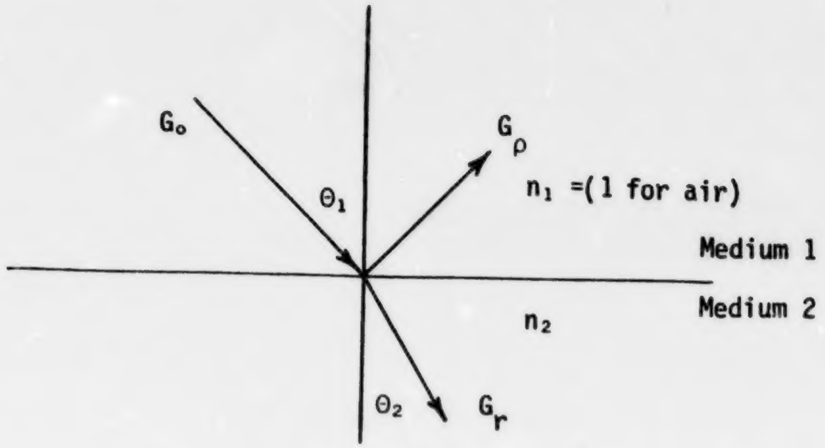


FIGURE 3.2.1

REFLECTION AT A SURFACE

The reflectance of the surface is a function of n_1, n_2, & Θ_1. Fresnel's law, which describes this relationship, is

$$r(o) = \frac{G_r}{G_i} = \left(\frac{n_1 - n_2}{n_1 - n_2}\right)^2 \qquad \text{(Eqn 3.2.2)}$$

for Θ, = 0, the relationship simplifies to

$$r(o) = \left(\frac{n - 1}{n + 1}\right)^2 \qquad \text{(Eqn 3.2.3)}$$

where the upper medium has $n = 1$ (a good approximation for air)

For glass, $n = 1.526$ (averaged over the solar spectrum) and

$$r(o) = \left(\frac{.526}{2.526}\right)^2 = .0434$$

For an incident angle of 60° from the normal, Fresnel's equation would yield

$$r(60) = 0.93$$

A cover glass has two surfaces, both reflecting. For normal incidents the reflectances are the same and the reflected and transmitted beam strengths arc as shown in Figure 3.2.2.

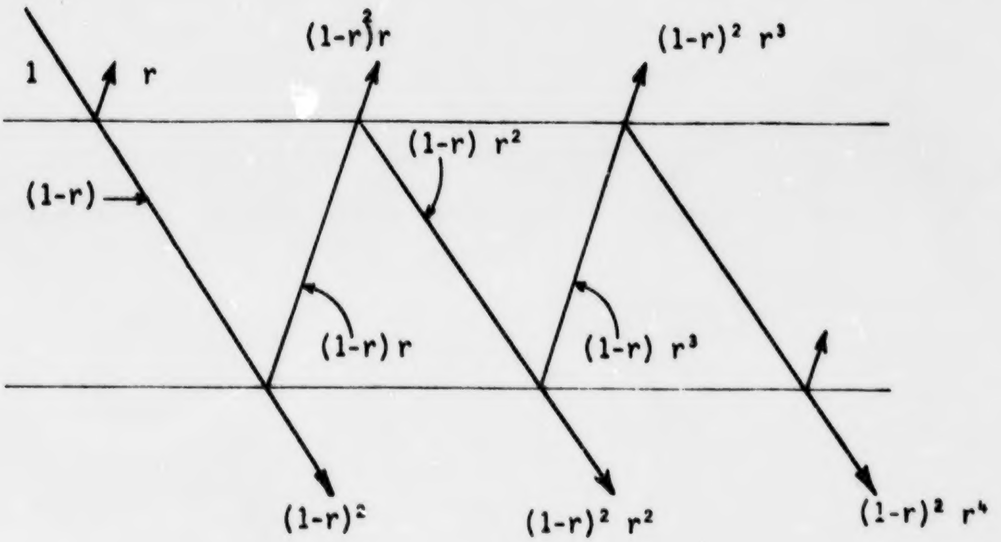

1
r
(1-r)²r
(1-r)² r³
(1-r) r²
(1-r)
(1-r) r
(1-r) r³
(1-r)²
(1-r)² r²
(1-r)² r⁴

FIGURE 3.2.2

TRANSMISSION THROUGH

ONE NON-ABSORBING COVER

Summing the transmitted terms yields

$$\tau_r = (1-r)^2 \sum_{j=o}^{\infty} r^{2j} = \frac{1 - r}{1 + r} \qquad \text{(Eqn 3.2.4)}$$

with N collector covers

$$\tau_{r,N} = \frac{1-r}{1+(2N-1)\,r} \qquad \text{(Eqn 3.2.5)}$$

for normal incidence.

Figure 3.2.3 shows how the transmittance of glass covers varies for other angles of incidence.

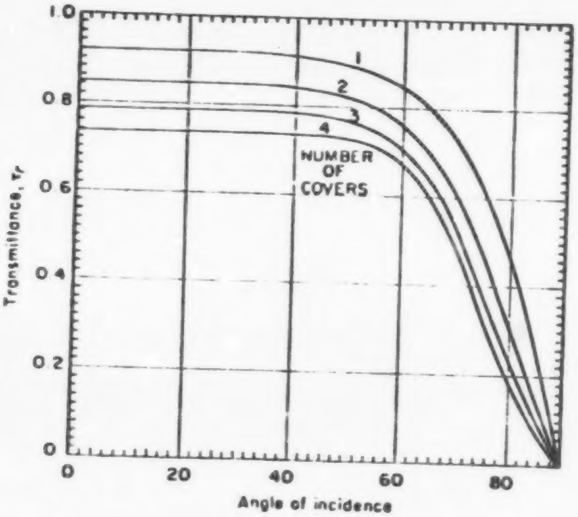


FIGURE 3.2.3

TRANSMITTANCE OF 1,2,3, AND 4 NONABSORBING COVERS HAVING AN INDEX OF REFRACTION OF 1.526

This analysis assumed no absorbtion in the glass. In fact, the beam strength will decay exponentially and the transmittance considering only absorbtion will be

$$\tau_\alpha = e^{-KL} \qquad \text{(Eqn 3.2.6)}$$

where L = path length and K = extinction coefficient

For glass $.04 < K < .32 \text{ cm}^{-1}$.

As an example, for high quality 0.32 cm (1/8") glass,

$$\tau_\alpha = e^{.04x.32} = .987$$

When $\tau_\alpha \cong 1$, the strengths of the beams in Figure 3.2.2 are only slightly affected by absorbtion, so that the two transmittances can be considered essentially independent and

$$\tau \cong \tau_\alpha \tau_r \qquad \text{(Eqn 3.2.7)}$$

This is a good approximation for collector cover plates. Using the values from our previous examples, a pane of highest quality glass 0.32 cm (1/8") thick will have a transmittance of

$$\begin{aligned} \tau &= \tau_\alpha \tau_r = \tau_\alpha \left(\frac{1-r}{1+r}\right) \\ &= .987 \left(\frac{1-.0434}{1+.0434}\right) \\ &= .905 \end{aligned}$$

for radiation at normal incidence. Table 3.2.1 has other examples.

Figure 3.2.4 shows the transmittances of several plate systems as a function of incident angle.

3.2.2 Absorbtion At The Plate Absorber

If an incident beam of intensity G falls on a solar collector, $I\tau$ will penetrate the cover system and $G\tau\alpha$ will be absorbed. However, the radiation reflected by the absorber is not lost, and may be recovered in subsequent reflections, as shown in Figure 3.2.5.

Table 3.2.1

GLAZING MATERIAL COMPARISON

Property	Teflon	Tedlar	Mylar	Lexan	Kalwal	PPG Herculite	ASG Tempered
Solar transmission (%)	96	92-94	85	82-89	85.90	85	91
Thickness (mm)	.025	.101	.356	3.175	.635	3.175	3.175
Tensile modulus (M Pa)	. .	17.7	37.4	23.5	74.8	680	680
Thermal expansion (%/° C x 10^3)	. .	4.32	2.70	6.75	3.60	1.55	1.55
Life (yrs)	15 (tested)	5 (5% loss)	4	. .	Indef. (93° C causes 10% loss)	Indef.	Indef.

** Data not available

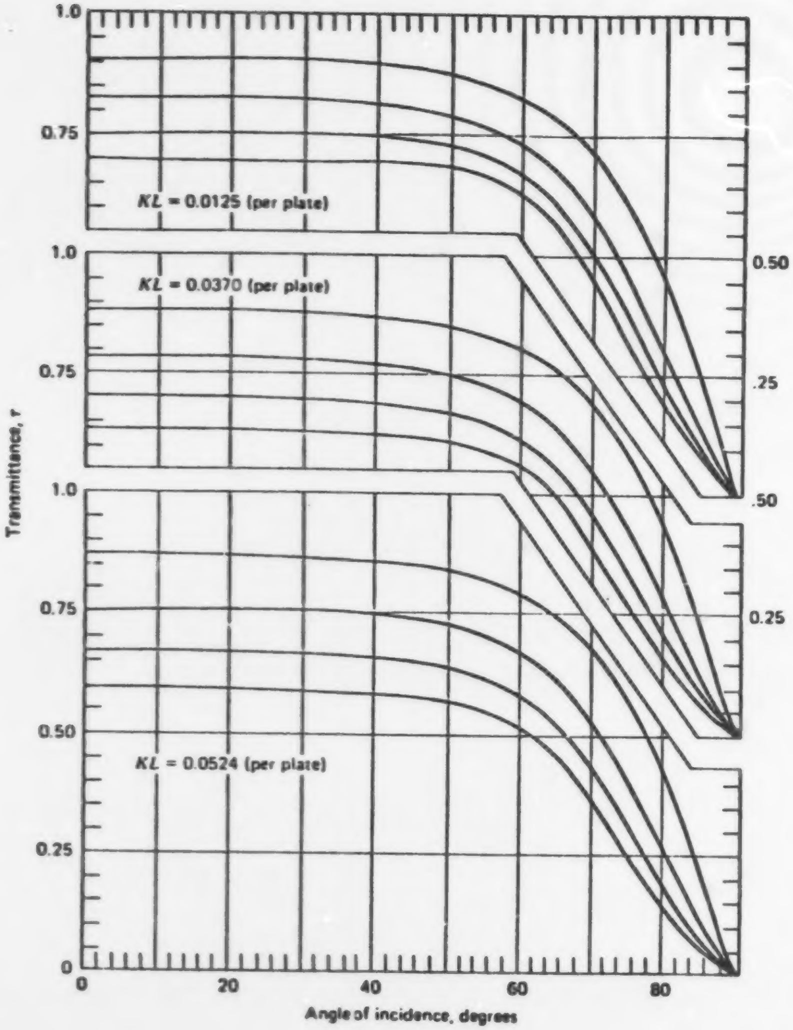


FIGURE 3.2.4

TRANSMITTANCE (CONSIDERING ABSORPTION AND REFLECTION)

OF 1,2,3 AND 4 COVERS FOR THREE TYPES OF GLASS

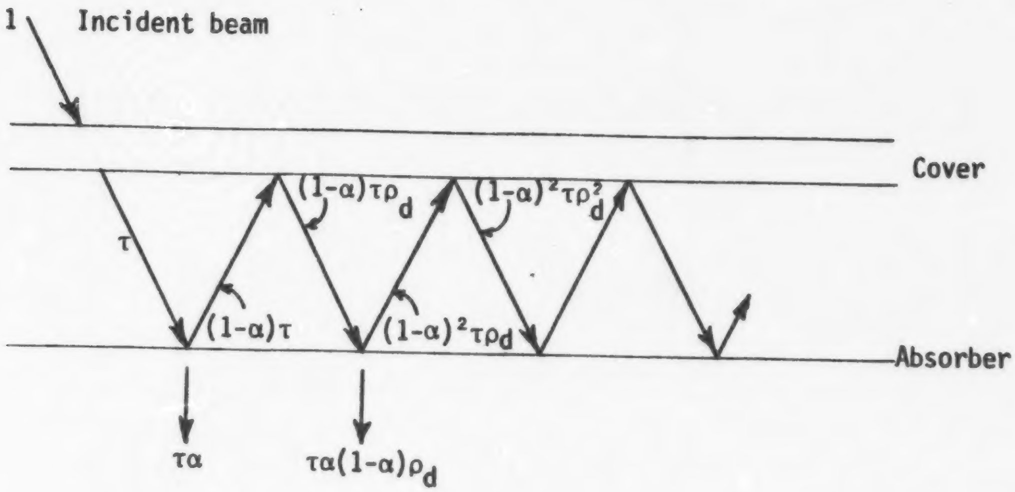

1
Incident beam
Cover
τ
$(1-\alpha)\tau\rho_d$
$(1-\alpha)^2\tau\rho_d^2$
$(1-\alpha)\tau$
$(1-\alpha)^2\tau\rho_d$
Absorber
$\tau\alpha$
$\tau\alpha(1-\alpha)\rho_d$

FIGURE 3.2.5
INTERACTION BETWEEN
COVER AND ABSORBER

Summing the absorbtion terms, the actual fraction of the incident beam absorbed is

$$(\alpha\tau) = \alpha\tau \sum_{j=o}^{\infty} \left[(1-\alpha)\ \rho_d\right]^j$$

$$= \frac{\alpha\tau}{1-(1-\alpha)\ \rho_d} \qquad \text{(Eqn 3.2.8)}$$

where ρ_d is the reflectance of the cover plates for diffuse radiation from below, and is approximately equal to ρ (60°).

The absorbtances of several surface coatings are given in Table 3.2.2.

Example 3.1

Find $(\alpha\tau)$ for two plates of the high quality glass from the previous two calculations over an absorber painted with 3M Black Velvet paint.

Table 3.2.2

ABSORBER SURFACE MATERIAL COMPARISON

Material	Solar Absorptance	Infrared Emittance	Absorptance Divided By Emittance
3M Black Velvet	.98	.90	1.1
Cat-a-loc flat black	.97	.86	1.1
Sicon flat black	.96	.83	1.2
Parson's black	.98	.92	1.1
Black Chrome Duramir BK	.97	.73	1.3
Nickel black on polished steel	.92	.11	8.4
Nickel black on galvanized iron	.89	.12	7.4
Nickel black, 2 layers on electroplated nickel on steel	.94	.07	13.0
CuO on nickel	.81	.17	4.8
Co_3O_4 on silver	.90	.27	3.3
CuO on aluminum	.93	.11	8.5
Cu Black on copper	.89	.17	5.2
Ebanol C on copper	.90	.16	5.6
CuO on anodized aluminum	.85	.11	7.7
PbS crystals on aluminum	.89	.20	4.5

Solution

For two cover plates

$$\tau_{r,2} = \frac{1-r}{1+(2N-1)r}$$

$$= \frac{1-.0434}{1+(4-1).0434}$$

$$= .846$$

$$\tau \quad = \tau_{r,2}\ \tau_{\alpha}^{\ 2}$$

$$= .846 \times (.987)^2$$

$$= .825$$

$\alpha \quad = 0.98$ (from Table 3.2.2)

$\rho_d \ \cong \rho\ (60°)$

and $\rho \ \cong \tau_{\alpha} - \tau$

From Figure 3.2.4, at KL = .0128 and at 60° incidence, τ=.73 for a two cover system.

$$\tau_{\alpha} = \varepsilon^{-KL} \qquad \text{(Eqn 3.2.6)}$$

$= .99$ Therefore $\rho_d = .99-.73$

$= .26$

$$(\alpha\tau) = \frac{\alpha\tau}{1-(1-\alpha)\ \rho_d} \qquad \text{(Eqn 3.2.8)}$$

$$= \frac{.98 \times .825}{1-(1-0.98).26}$$

$$= .813$$

3.3 HEAT TRANSFER IN FLAT PLATE COLLECTORS

3.3.1 Introduction

The useful energy collected by a flat plate collector is

$$Qu = A_c \left[S - U_L \left(T_{p,m} - Ta \right) \right] \qquad \text{(Eqn 3.3.1)}$$

where S = the energy absorbed/unit area

$= (\alpha\tau)\ I_T$

U_L = the overall heat loss coefficient

$T_{p,m}$ = the mean plate temperature

Ta = the ambient temperature

and A_c = the collector area.

The main problem of our analysis of collector heat transfer will be to modify this equation so that it can be written in terms of the known inlet temperature instead of the unknown mean plate temperature. Figure 3.3.1 shows the temperature distribution on the absorber plate. The temperature varies in a complex way both along and across the plate. From

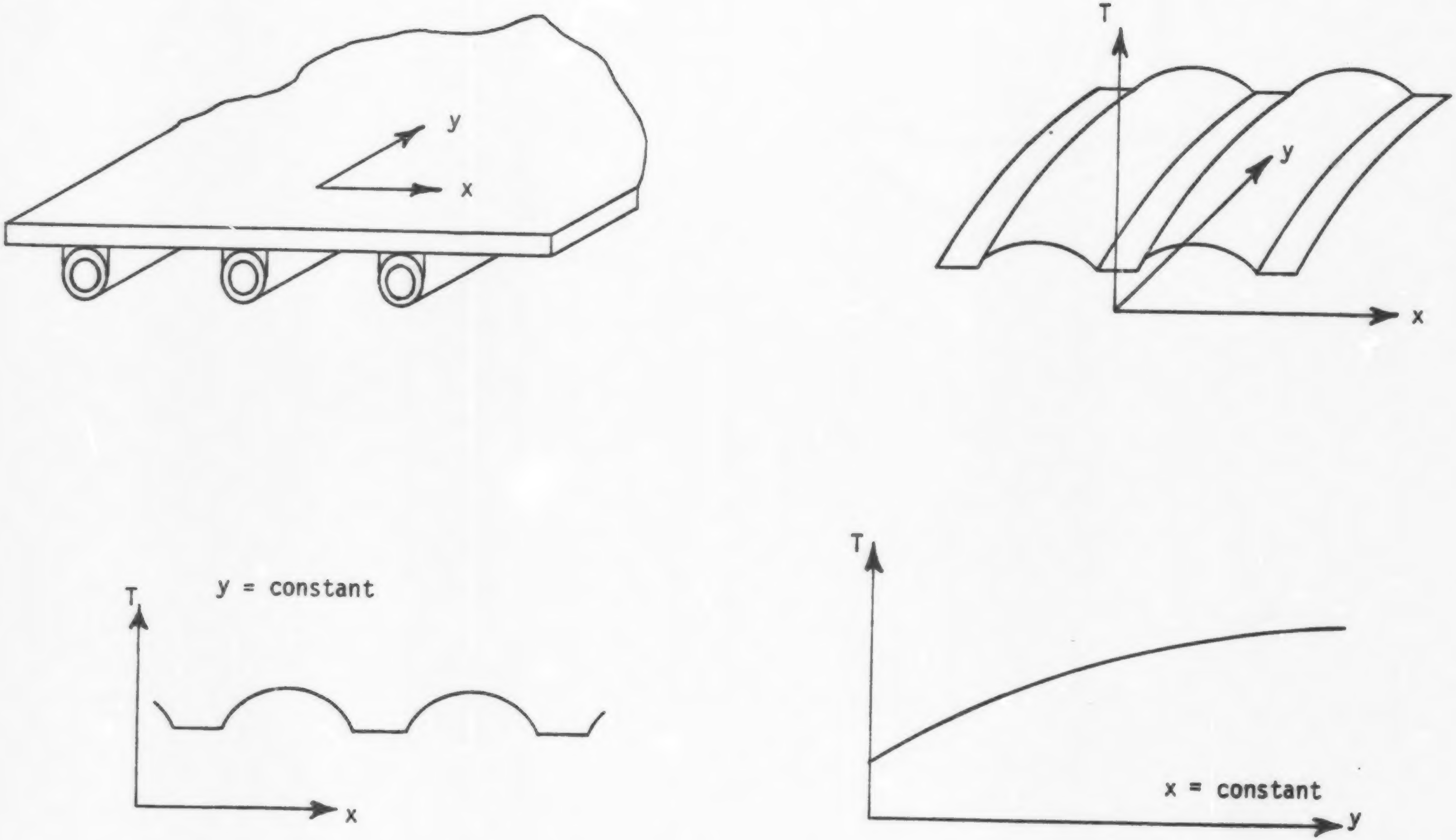
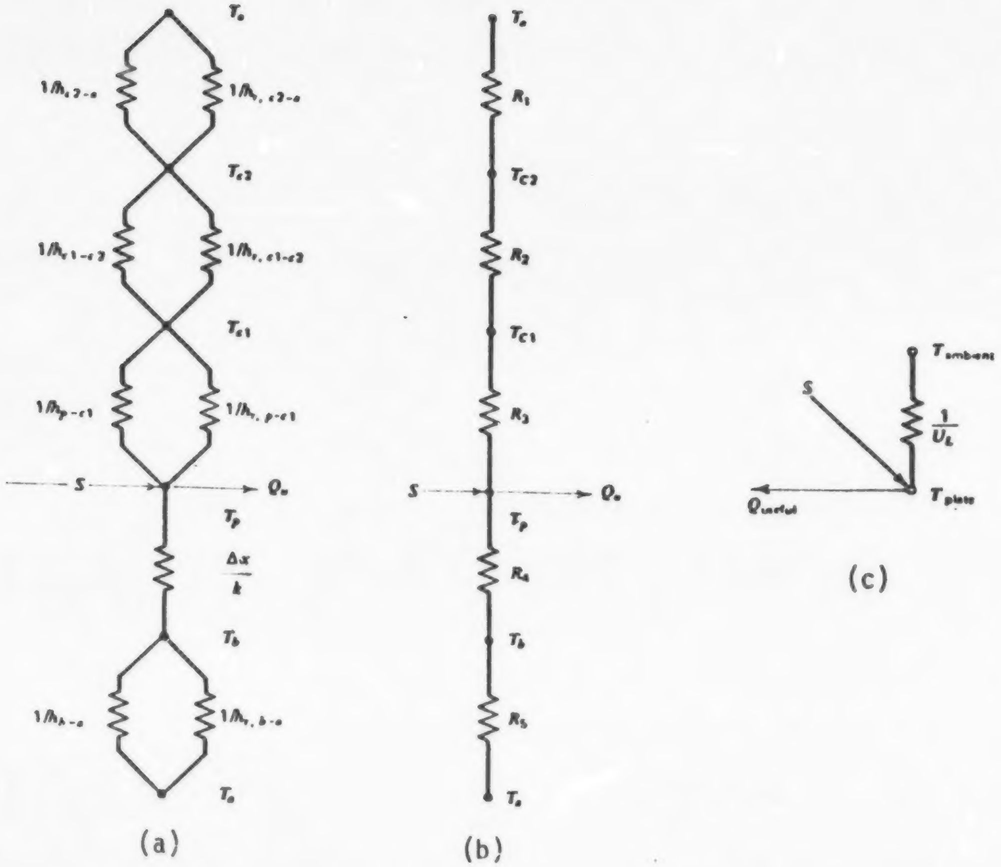

y
x
T
y
x
T
y = constant
x
T
x = constant
y

FIGURE 3.3.1

TEMPERATURE DISTRIBUTION ON ABSORBER PLATE


T_a
$1/h_{c2-a}$
$1/h_{r,c2-a}$
T_{c2}
$1/h_{c1-c2}$
$1/h_{r,c1-c2}$
T_{c1}
$1/h_{p-c1}$
$1/h_{r,p-c1}$
S
Q_u
T_p
$\frac{\Delta x}{k}$
T_b
$1/h_{b-a}$
$1/h_{r,b-a}$
T_a
T_a
R_1
T_{C2}
R_2
T_{C1}
R_3
S
Q_u
T_p
R_4
T_b
R_5
T_a
$T_{ambient}$
S
$\frac{1}{U_L}$
T_{plate}
Q_{useful}

(a) (b) (c)

FIGURE 3.3.2

THERMAL NETWORK FOR A TWO COVER

FLAT-PLATE COLLECTOR

(a) In terms of Conduction Convection, and Radiation resistances,

(b) In terms of resistances between plates, and

(c) In terms of overall loss coefficient this it can be seen that the mean plate temperature would be very difficult to measure or predict for a particular operating condition.

3.3.2 The Collector Heat Loss Coefficient

The radiant heat transfer between two parallel plates is given by

$$\frac{Q_r}{A} = \frac{\sigma\,(T_2^4 - T_1^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad \text{(Eqn 3.3.2)}$$

We can define a radiation heat transfer coefficient

$$h_r = \frac{\sigma(T_2 + T_1)\,(T_2^2 + T_1^2)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad \text{(Eqn 3.3.3)}$$

so that $Q_r = h_r A\,(T_2 - T_1)$ (Eqn 3.3.4)

For radiant interchange with the sky

$$Q_r = \varepsilon A \sigma\,(T^4_{sky} - T^4) \qquad \text{(Eqn 3.3.5)}$$

With $h_r = \dfrac{\varepsilon\sigma(T_{sky} + T)\,(T^2_{sky} + T^2)(T - T_{sky})}{(T - T_{amb})}$ (Eqn 3.3.6)

we can write

$$Q_r = h_r\,A\,(T - T_{amb})$$

Note that these radiant heat transfer coefficients are not constants. They vary with the temperatures of the radiating bodies. Of course convective heat transfer coefficients also vary with temperature, but not nearly as much as radiative coefficients. Therefore, where radiation is an important mechanism of heat transfer, it is unlikely that heat flux will vary linearly with temperature difference over a wide range of temperatures.

The thermal resistances in a solar collector can be characterised as shown in Figure 3.3.2(a) and simplified as shown in Figures 3.3.2(b) and 3.3.2(c).

Thus $U_L = U_t + U_b$

where $U_t = \frac{1}{R_1 + R_2 + R_3}$ (Eqn 3.3.7)

and $U_b = \frac{1}{R_4 + R_5}$

with each of the thermal resistances which represents parallel radiation and convection given by

$$R = \frac{1}{h_r + h_c}$$

D and B present, on Pp 207-209, a series of graphs like those shown in Figure 3.3.3 for the top loss coefficients for various situations.

Figure 3.3.4 shows the magnitudes of the heat flows and cover plate temperatures for the following circumstances

$T_{plate} = 100°C$

$T_{amb} = T_{sky} = 10°C$

Plate spacing = 25 cm

Tilt = 45°

$h_{wind} = 10W/m^2\ °C$

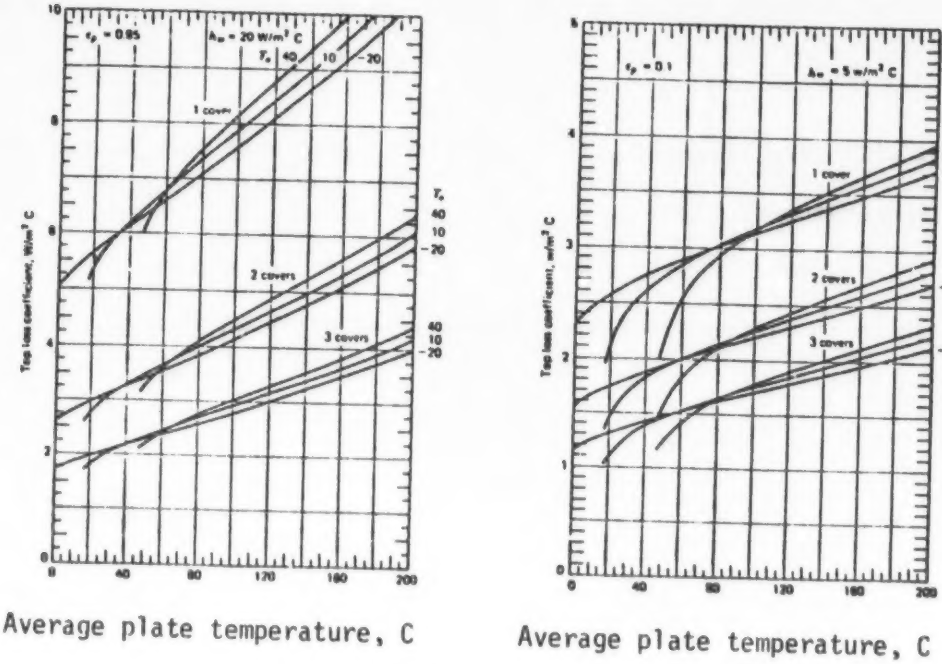


FIGURE 3.3.3

TOP LOSS COEFFICIENTS vs PLATE TEMPERATURE

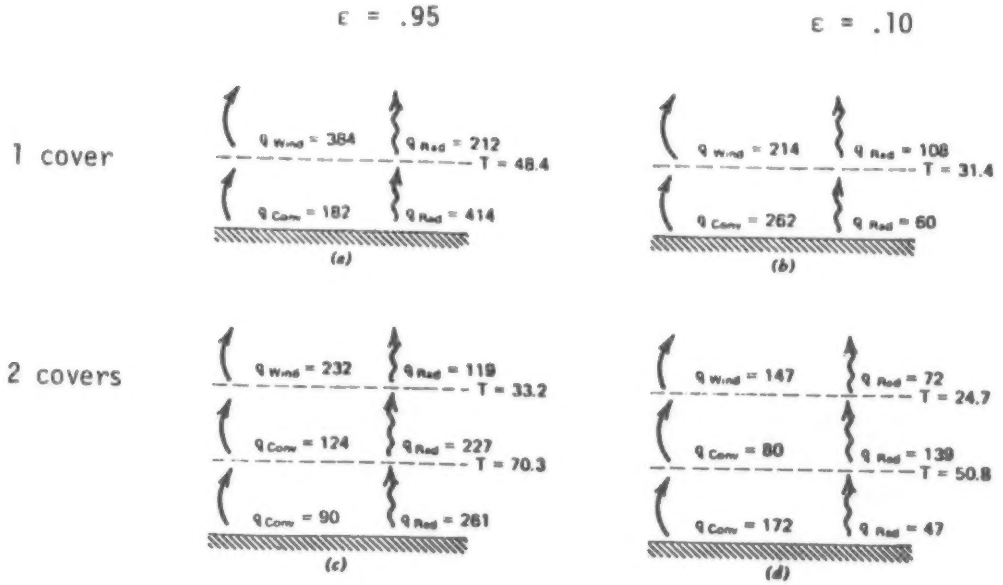

ε = .95
ε = .10
1 cover
q Wind = 384
q Rad = 212
T = 48.4
q Conv = 182
q Rad = 414
(a)
q Wind = 214
q Rad = 108
T = 31.4
q Conv = 262
q Rad = 60
(b)
2 covers
q Wind = 232
q Rad = 119
T = 33.2
q Conv = 124
q Rad = 227
T = 70.3
q Conv = 90
q Rad = 261
(c)
q Wind = 147
q Rad = 72
T = 24.7
q Conv = 80
q Rad = 139
T = 50.8
q Conv = 172
q Rad = 47
(d)

FIGURE 3.3.4

TYPICAL HEAT FLOWS

AND PLATE TEMPERATURES

Figure 3.3.5 shows the effects of plate spacing, absorber emittance and absorber temperature on U_L. It is interesting to note that absorber temperature has a substantial effect on U_L, even for a selective surface where radiation losses are minimized. Also notice that U_L is at a minimum at a cover plate spacing between 10 and 15 cm.

3.3.3 Temperature Distribution On The Plate Of A Conventional Liquid Cooled Collector

3.3.3.1 Temperature Distribution Across The Plate

The absorber plate is a fin transferring heat to the coolant tubes, and its thermal performance is like that of any other fin. The fin efficiency is defined as

F = (actual heat gain)/(heat gain if the whole fin were at the base temperature)

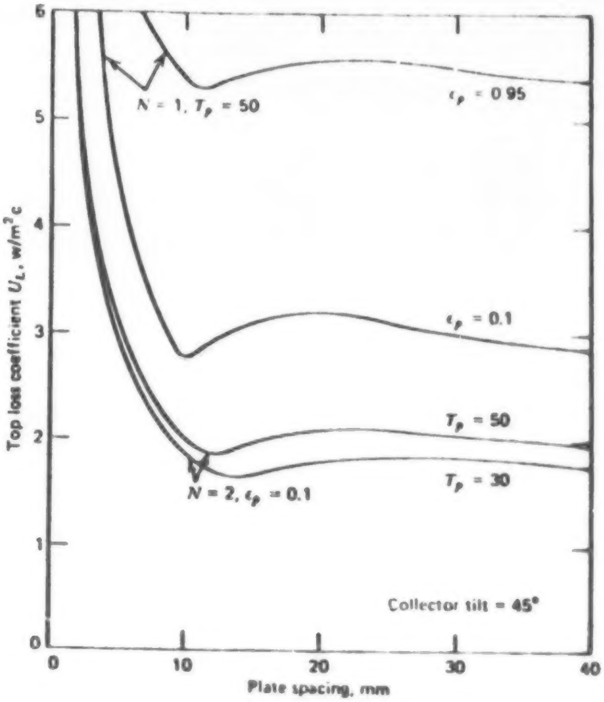


FIGURE 3.3.5

VARIATION OF TOP LOSS COEFFICIENT WITH PLATE SPACING

F approaches 1.0 as the fin gets shorter or thicker or as the thermal conductivity approaches infinity.

Using the notation of Figure 3.3.6, the fin efficiency is given by

$$F = \frac{\tanh\,[m(W-D)/2]}{m(W-D)/2} \qquad \text{(Eqn 3.3.8)}$$

where $m = (U_L/k\delta)^{0.5}$

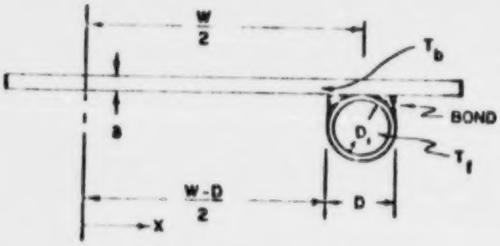


FIGURE 3.3.6

NOMENCLATURE FOR FIN ANALYSIS

Figure 3.3.7 shows the variation of F with m(W-D)/2.

The collector efficiency factor F' is defined as the actual useful energy gain divided by the energy gain if the absorber plate were at the fluid temperature. The collector efficiency factor is given by

$$F' = \frac{1/U_L}{W\left[\dfrac{1}{U_L[D+(W-D)F]} + \dfrac{1}{C_b} + \dfrac{1}{\pi D_i h_{f,i}}\right]} \qquad \text{(Eqn 3.3.9)}$$

C_b is the bonding coefficient between the absorber and the coolant tubes (in units of W/°C per meter of the tube length) so that $1/C_b$ is the plate to tube thermal resistance per unit length of tube. $h_{f,i}$

is the convective heat transfer coefficient inside the tubes, so that $\frac{1}{\pi D_i h_{f,i}}$ is the tube to fluid thermal resistance per unit length of tube. $[D+(W-D)F]$ is the tube pitch, corrected by the fin efficiency F for the fact that there is a thermal resistance between some parts of the plate and the place where they bond to the tubes. Thus $\frac{1}{U_L[D+(W-D)F]}$ is the thermal resistance between the environment and

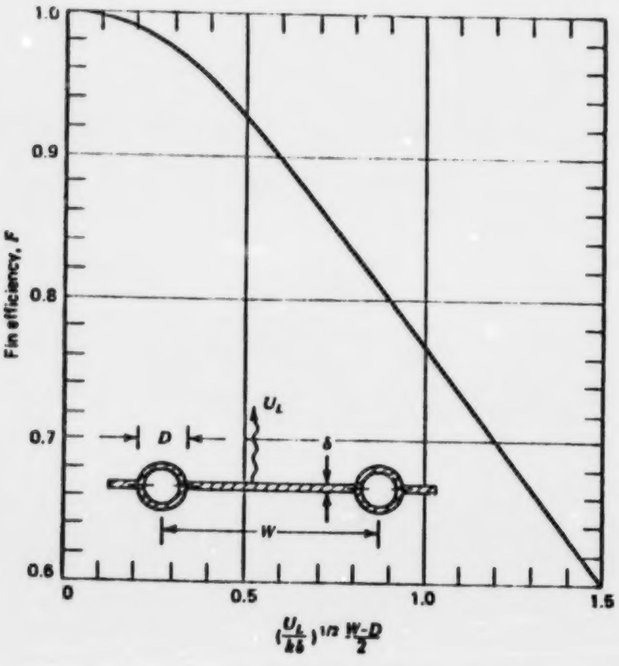


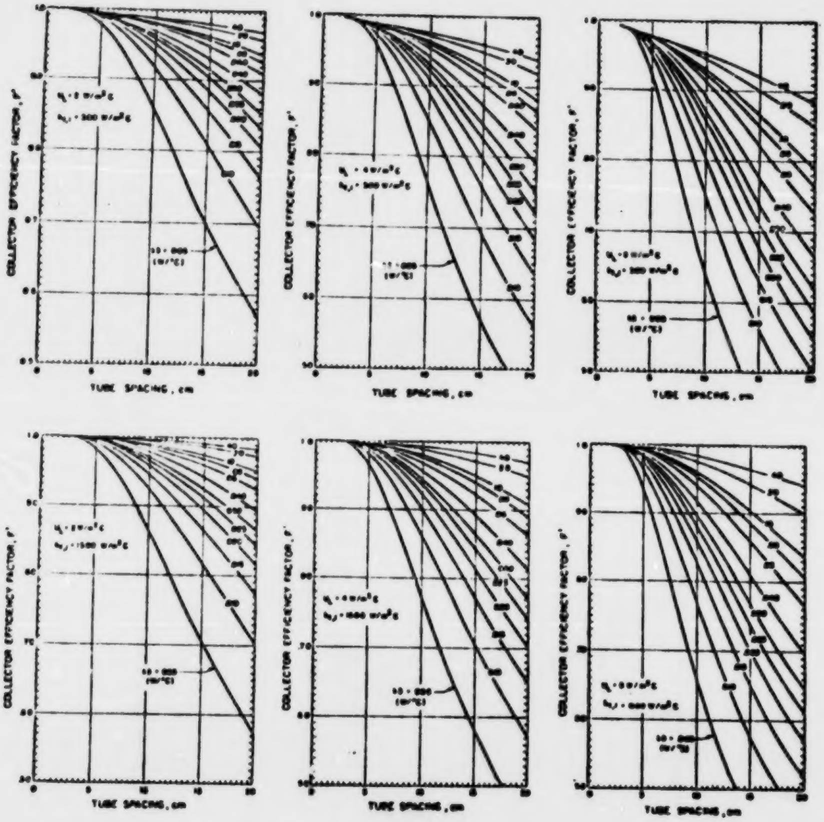
FIGURE 3.3.7

VARIATION OF FIN EFFICIENCY

WITH M(W-D)/2

the place where the plate bonds to the tubes (per unit length of tube) Thus F' = R_L Ro where R_L is the thermal resistance from the plate to the environment and Ro is the thermal resistance from the tube to the environment.

Figure 3.3.8 shows the variation of F' with tube spacing, U_L, $h_{f,i}$ and kδ for 20 mm diameter tubes. Duffie and Beckman present similar data for 10 mm tubes in Figure 6.5.4, Pp 219-221.



Collector efficiency factor F' versus tube spacing for 2 cm diameter tubes for various conditions.

FIGURE 3.3.8

3.3.3.2 Temperature Distribution Along The Plate

The thermal performance of a solar collector is decreased not only because the plate temperature increases between coolant tubes, but also because the fluid temperature, and thus the plate temperature, increases from the inlet to the outlet. As the fluid temperature increases, the local efficiency and thus the rate of temperature increase, is lowered, resulting in a temperature profile like that shown in Figure 3.3.9.

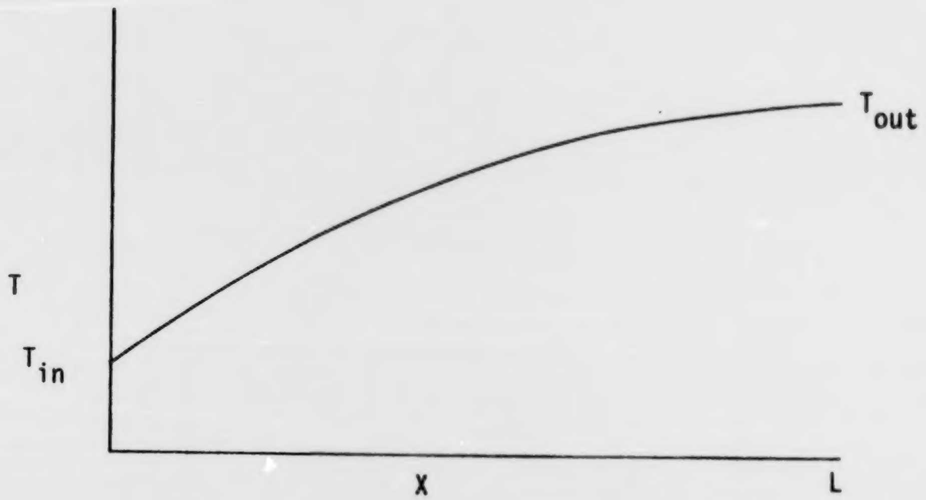


FIGURE 3.3.9

FLUID TEMPERATURE vs

POSITION ALONG TUBE

To account for this decrease in performance we introduce the flow factor

F" = (energy gain)/(energy gain if the fluid temperature were everywhere equal to the inlet temperature).

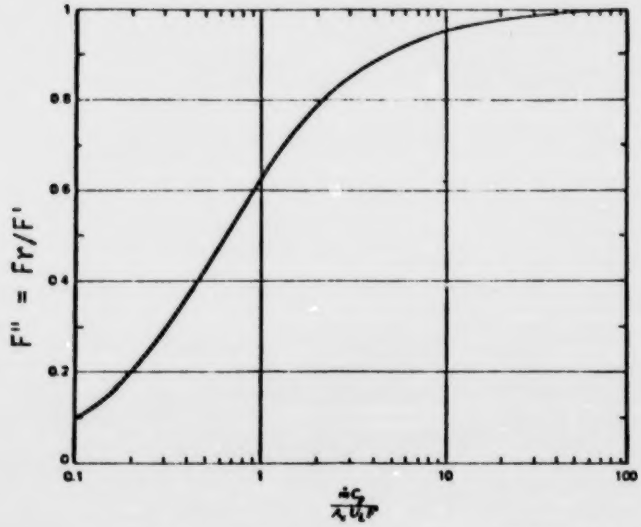
Obviously, if the fluid thermal capacity rate, $\dot{m}C_p$, were very large, F" would approach 1.0.

It can be shown (see D and B Sec 6.7) that

$$F'' = \frac{1}{z}(1 - e^{-z}) \qquad \text{(Eqn 3.3.10)}$$

where $z = \dfrac{A_c F' U_L}{\dot{m}\, Cp}$

Figure 3.3.10 shows F" as a function of 1/z.



Collector flow factor F'' as a function of $\dot{m}C_p/A_c U_L F'$.

FIGURE 3.3.10

Combining the two previous concepts, it becomes possible to introduce the collector heat removal factor

$$F_R = F'F'' \qquad \text{(Eqn 3.3.11)}$$

This is the ratio of the useful energy gain to the useful energy gain which would occur if the entire plate were at the fluid inlet temperature. Combining equations 3.3.10 and 3.3.11

$$F_R = \frac{m\ Cp}{A_c U_L} \left(1 - e^{-A_c U_L F'/\dot{m}\ C_p}\right) \qquad \text{(Eqn 3.3.12)}$$

Introducing F_R into Eqn 3.3.1 yields

$$Qu = A_C\ F_R \left[S - U_L\ (T_{f,i} - Ta)\right] \qquad \text{(Eqn 3.3.13)}$$

where $S = G_T\ (\alpha\tau)$

However, there is a further correction to make to this equation. When $(\alpha\tau)$ was evaluated, the effect of the radiation absorbed by the cover plates on their temperature was ignored. This slight increase in temperature will decrease the losses from the absorber. This can be taken into account by introducing the effective transmittance-absorbtance product,

$$(\alpha\tau)_e = (\alpha\tau) + \left(1 - \tau_{a,1}\right) a_1$$
$$+ \left(1 - \tau_{a,2}\right) a_2 \tau_1 + \left(1 - \tau_{a,3}\right) a_3 \tau_2 + \cdots \qquad \text{(Eqn 3.3.13)}$$

where a_i is the ratio of the top loss coefficient to the loss coefficient from the i_{th} cover to the atmosphere. For example, for a single cover of low iron glass over a black painted surface

$$(\alpha\tau)_e = (\alpha\tau) + \left(1 - \tau_a\right) a_1$$
$$= (.98 \times .906) + (1 - .988)\ .427$$
$$= .888 + .005$$
$$= .893$$

In this case, the correction is of the same magnitude as the errors in the property data. Only in double glazed collectors with cover plates having high absorbtion does this correction become significant.

Equation 3.3.13 can now be written as

$$Q_u - A_C F_R G_G (\alpha\tau)_e - A_C F_R U_L (T_{f,i} - T_a) \qquad \text{(Eqn 3.3.14)}$$

3.3.4 Other Collector Geometries

The analysis of Section 3.3.3 applies to a liquid cooled tube in plate type collector of conventional design. A similar approach has been taken for many other geometries. It is usually possible to express the results in a similar form, with slightly different expressions for F' and/or F_r. D and B present the results of several such analyses in Section 6.12, beginning on Pg. 236.

A geometry which is used in one of the PUSH approved liquid cooled collectors is the serpentine tube. D and B, in Example 6.12.1 presents an analysis of F_r for a serpentine tube collector. A parallel tube collector of the same size, materials and mass flow rate would have about the same value of F_r, but the pressure drop to maintain this flow rate would be considerably less.

In general, air cooled collectors will have lower values of F_r than liquid cooled collectors, because of the lower specific heat, lower mass flow rate and lower heat transfer coefficient. However, an air cooled collector using a porous absorber plate through which the air is drawn can have a value of F_r very close to 1.0. Such a collector is shown in Figure 6.12. i, Pg. 239, D and B. One of the PUSH approved air cooled collectors uses this geometry.

3.4 FLAT PLATE COLLECTOR THERMAL PERFORMANCE

ASHRAE Standard 93-77 prescribes the standard method of test for rating the thermal performance of flat plate collectors. (Note that this standard is presently undergoing revision and a new version will be published in 1983.) This standard test method is based on the heat transfer analysis of Section 3.3.

The efficiency of a flat plate collector is defined as

$$\eta = Q_u \; A_c G_t \qquad \text{(Eqn 3.4.1)}$$

Substituting Q from equation 3.3.14

$$\eta = F_R(\alpha\tau)_e - F_R U_L \left(\frac{T_{f,i} - T_a}{G_t} \right)$$

$F_R(\alpha\tau)_e$ and $F_R U_L$ are properties of the collector. They vary only slightly with temperature. $\left(T_{f,i} - T_a\right)/G_t$ depends only on conditions external to the collector. Therefore, for any one collector, the effiency can be plotted in the form $\eta = a + bx$ with the y-intercept $a = F_R(\alpha\tau)_e$ and the slope $b = F_R U_L$.

The abscissa $(T_{f,i} - T_a)/G_t$ combines all the external variables into a single term.

Figure 3.4.1 shows a typical ASHRAE 93-77 efficiency plot, prepared by fitting a straight line through four sets of four data points, measured over a wide range of values of $\left(T_{f,i} - T_a\right)/G_t$. It is important to note that the efficiency curve is based on gross area, so that losses due to radiation falling on the collector frame instead of the cover plate lower the apparent collector efficiency. The ASHRAE 93-77 collector thermal performance tests are carried out with radiation falling normally on the collector, so the "incident angle modifier" must also be measured, so that the higher reflective losses at incident angles away from normal can be accounted for. Figure 3.4.2 shows a typical incident angle modifier curve.

The Incident Angle Modifier is defined as:

$$K_{\alpha\tau} = (\alpha\tau) / (\alpha\tau)$$

so that the useful energy collected is then given by

$$Qu = A_c F_R \left[G_T K_{\alpha\tau} (\alpha\tau)_n - U_L (T_i - T_a)\right]$$

The variation of $K_{\alpha\tau}$ with incident angle, θ, can be approximated by the expression

$$K_{\alpha\tau} = 1 + b_o \ (1/\mathrm{Cos}\ \theta - 1)$$

where b_o is a constant, called the incident angle modifier coefficient, which can be determined experimentally for each collector.

The Canadian Solar Industries Association publishes descriptions of all PUSH approved solar collectors. On the last page of each collector description are the results of an ASHRAE 93-77 test of the collector. The results shown are only valid for the same heat transfer fluid at the same flow rate. UNIES Ltd. has prepared a set of graphs for the Solar Programs Office which will be published shortly and which will provide correction factors for the common fluids over a range of flow rates.

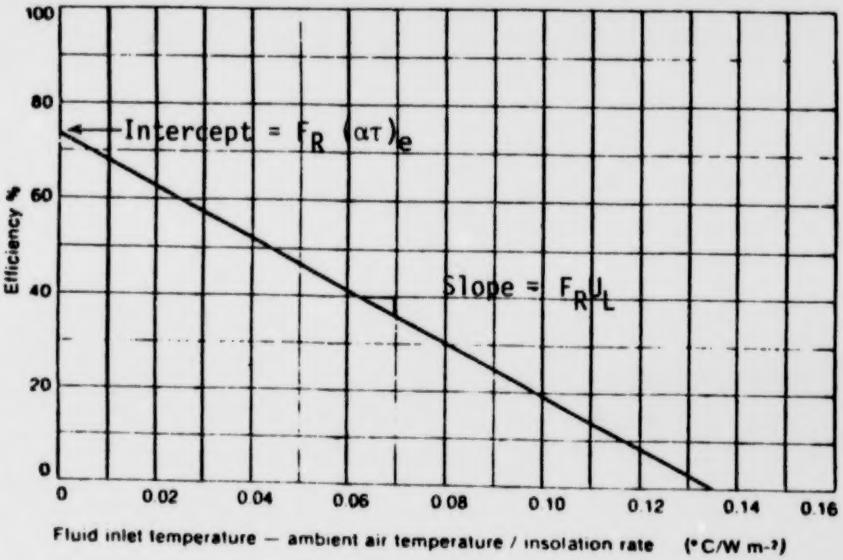


FIGURE 3.4.1

COLLECTOR EFFICIENCY CURVE

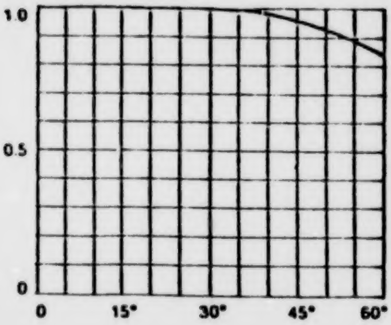


FIGURE 3.4.2

INCIDENT ANGLE MODIFIER

Example

A solar collector has a copper absorber 0.5 mm thick, coated with 3M Black Velvet paint. It is double glazed with water white (low iron) glass. The coolant tubes are 2 cm. in diameter and are spaced at 15 cm. intervals. The collector is mounted at 45°. The wind heat transfer coefficient is 10 W/m² °C. The ambient temperature is 10°C, the fluid inlet temperature is 40°C, the coolant is water flowing at 3.5ℓ/min. The heat transfer coefficient inside the tubes is 1500W/m² °C, and the collector area is 1.7 m². The back loss coefficient is 1.0 W/m² °C. Find the expected values of $F_R(\alpha\tau)$ and F_RU_L.

Solution

From previous examples, for this absorber-cover combination $(\alpha\tau) = .0.82$.

For copper, $K = 386$ W/m °C. ∴ $K\delta = 386 \times .0005 = 0.193$ W/°C.

From Figure 6.4.4(b), Pg 207, D and B, $U_{TOP} = 3 \text{W/m}^2$ °C.

Therefore $U_L = U_{TOP} + U_{BACK}$

$$= 3 + 1$$

$$= 4.0 \text{ W/M}^2 \text{ °C}$$

$$\text{and} \left(\frac{U_L}{K\delta}\right)^{\frac{1}{2}} \frac{W-D}{2} = \left(\frac{4}{.193}\right)^{\frac{1}{2}} \times \frac{.13}{2}$$

$$= .2959$$

From Figure 3.3.7, F = .9718

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L(D+(W-D)F} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{f,i}}\right]}$$

$$= \frac{1/4}{.15\left[\frac{1}{4\ (.02 + .13 \times .9718)} + 0 + \frac{1}{\pi \times .02 \times 1500}\right]}$$

$$= \frac{.25}{.15\ [1.7084 + .0106]}$$

$$= .9695$$

$$\therefore \left(\dot{m}\ C_p/U_L F^1 A_c\right) = \frac{3500/60 \times 4.184}{4.0 \times .9695 \times 1.7}$$

$$= 37.02$$

From Figure 3.3.9, $F'' = .9866$

$\therefore F_R = F'F'' = .9695 \times .9866$

$= .957$

$\therefore F_R(\alpha\tau) = .957 \times .83$

$= .79$

and $F_R U_L = .957 \times 4.0$

$= 3.9 \text{ W/m}^2 \text{ °C}$

Assignment

A solar collector has an absorber plate of steel 1 mm thick. The abosrbing surface is nickel black. The tube spacing is 15 cm and the tube diameter is 2 cm. The collector has a single cover plate of .014" thick mylar. The mass flow rate of the cooling water is 2 ℓ/min and the heat transfer coefficient inside the tubes is 300 W/M² °C. The collector area is 1.7 m². Find F_R $(\alpha\tau)$ and $F_R U_L$.

3.5 COLLECTOR ARRAY PERFORMANCE

All the above analysis applies to individual collectors. When the collectors in an array are connected in parallel, each collector receives coolant at the same inlet temperature, so the same analysis is valid. When collectors are connected in series, so that the coolant which has been heated in one collector enters the next, this is not the case. Therefore the results obtained above must be modified to take this into account.

One PUSH approved collector manufacturer, Nortec Solar Industries Inc., provides a graph in their product information brochure which allows the collector performance to be corrected for series mounting. Where this information is not available, it is possible to correct the collector performance by assuming that two collectors in series will perform the same as a single collector with half the flow. Methods of carrying out this correction will be discussed in Part 2 of this course.

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CHAPTER 4

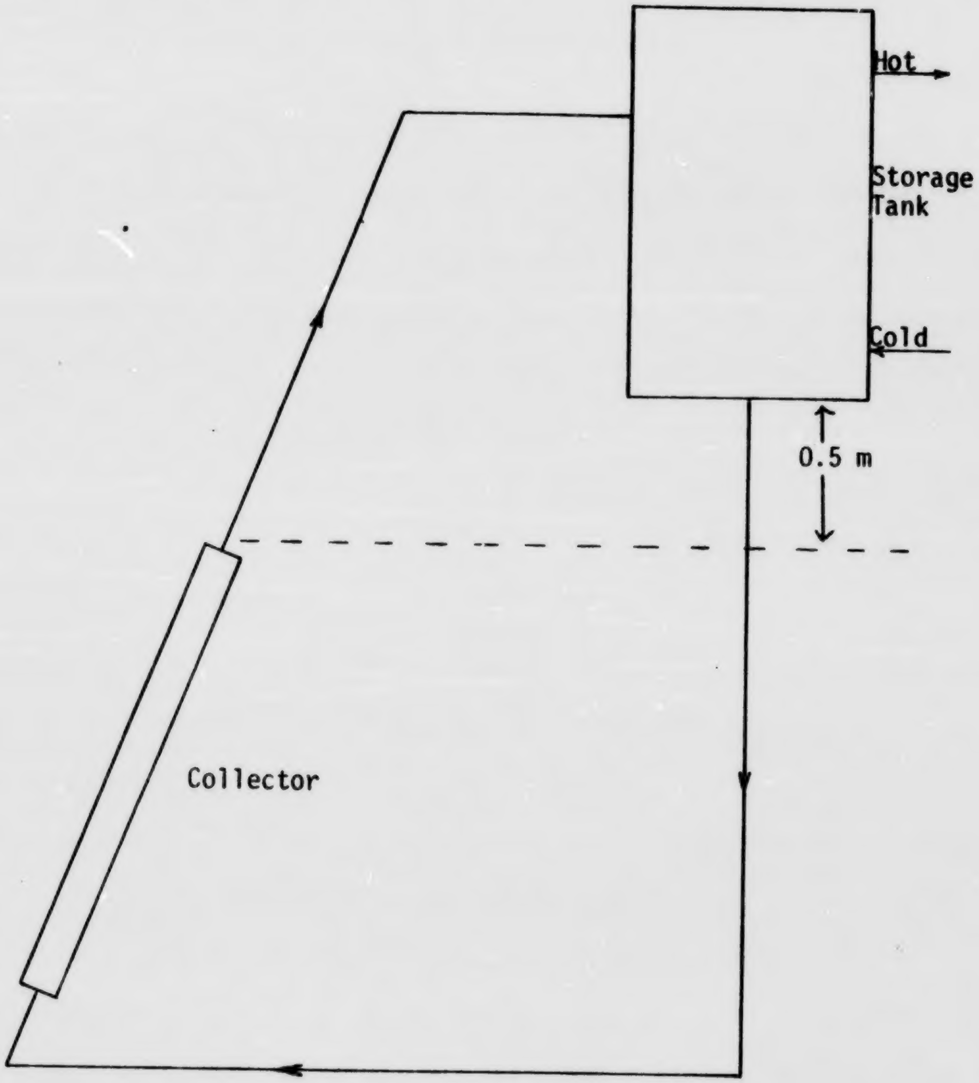
LIQUID SYSTEMS

The design of a solar energy system is dependent on use of the energy collected and on the physical and environmental conditions the systems must operate in. The most common uses made of solar energy are space heating, service water heating and swimming pool heaters. Solar cooling systems using absorption refrigerating cycles are commercially available, but due to their poor economics, are not discussed here.

4.1 TYPES OF SYSTEMS

The simplest use of solar energy is service water heating using a direct heating passive design. This type of system ranges from outdoor metal water storage tanks found throughout the tropics which may or may not be glazed to reduce thermal losses, to thermosiphon circulation systems. These systems are for non-freeze installations, so their use in Canada is limited to summer operation in all localities.

The thermosiphon circulation system (Figure 4.1.1) may operate in a pressure system, or in an open system. Solar radiation warms the water in the collectors. This water rises through the collector by convection into the top of the storage tank, which is located above the top of the collector. Cold water from the bottom of the storage tank is piped into the bottom of the collectors, displacing the warm outlet water. The bottom of the storage tank must be 0.5 m or more above the top of the collectors. This elevation difference, along with short, well insulated pipe runs en-


Hot
Storage
Tank
Cold
0.5 m
Collector

Direct Heating Thermosiphon Circulation Type of Solar Waste Heater

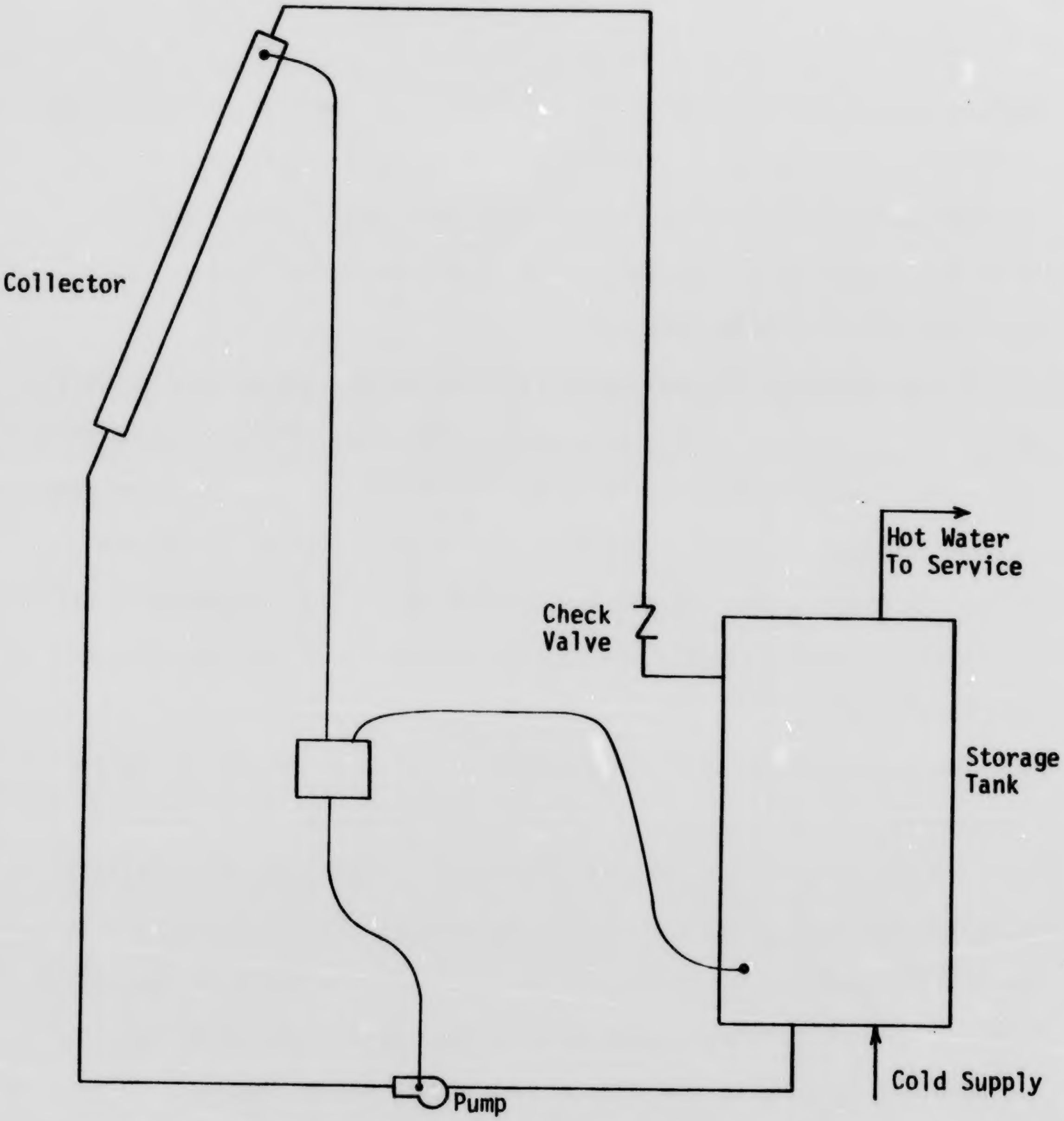
FIGURE 4.1.1

courages convective movement which will limit the temperature rise in the collectors, thus improving efficiency. If the solar collectors are not located completely below the storage tank, when the collector temperature falls below that of the storage, convection currents may reverse, and major heat losses can be incurred.

A high degree of thermal stratification in the storage tank results and so the collectors utilize only the coldest water from storage, with the highest temperature water being available for service. As solar radiation increases, so does the rate of circulation. ASHRAE comparison tests show thermosiphon systems are more efficient than mechanically driven circulation systems, largely due to required deadbands and inaccuracies in control settings.

The major problem with thermosiphon circulation type water heaters is the critical relationship of the relative locations of the storage tank and collectors. The simplest way to circumvent this is to install a circulation pump (Figure 4.1.2) in the circulation loop. Controls to operate the pump can consist simply of a temperature sensor in the collectors but for more efficient operation, a temperature comparator should be used with sensors in the storage tank and on the collectors. If the collectors are located above the storage tank, a check valve is required to prevent heat losses caused by the reverse thermosiphoning of stored water through the collectors when the pump is off.

These are direct heating systems. That is, water is heated directly in the collectors. Direct systems such as these require some form of


Collector
Check Valve
Hot Water To Service
Storage Tank
Cold Supply
Pump

Direct Heating, Pump Circulation Type Of Solar Water Heater

FIGURE 4.1.2

freeze protection in Canada. Drain back and drain-down systems can provide this protection by automatically draining all the water from the exposed system components when freezing temperatures are approached or when the system is not in operation. Fail safe designs are critical to long term system operation.

A drainback system (Figure 4.1.3) usually operates with an unpressurized thermal storage system. The solar fluid circulation lines are vented and graded so that the solar fluid will drain back to the thermal storage tank when the circulation pump is not operating. When the circulation pump starts, the system will be re-filled with solar fluid.

Advantages of drainback systems include:

(1) system has freeze protection

(2) controls are simple

(3) they can use water as the solar fluid. Water is cheap and doesn't degrade with use.

(4) being direct heating systems, they enjoy high operating efficiencies

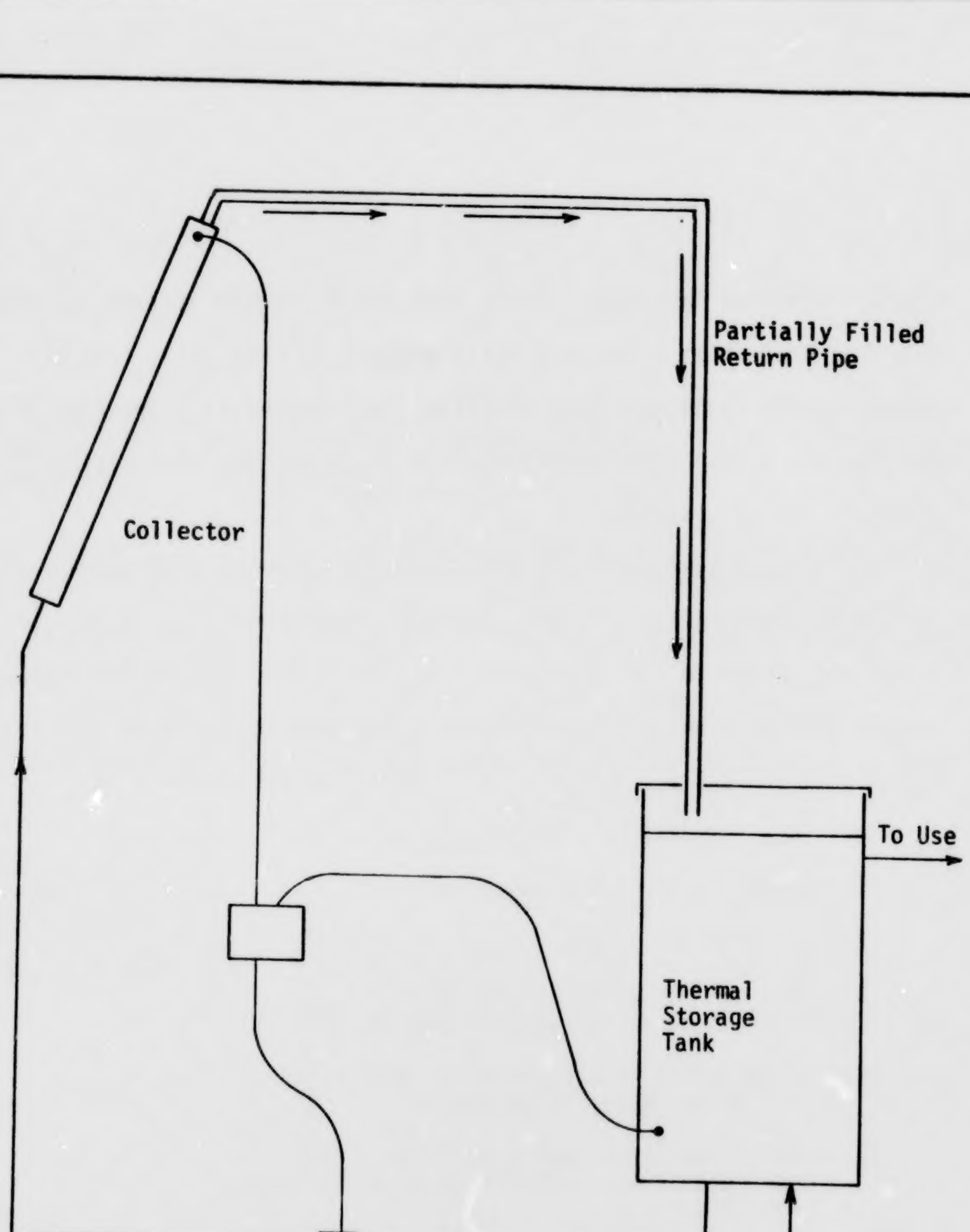
(5) the system is mechanically simpler and thus cheaper than other systems with freeze protection

(6) drainage of solar fluid from collectors to thermal storage results in recovery of solar energy which might otherwise be lost to the atmosphere when the system shuts down.

The disadvantages of drainback systems are:

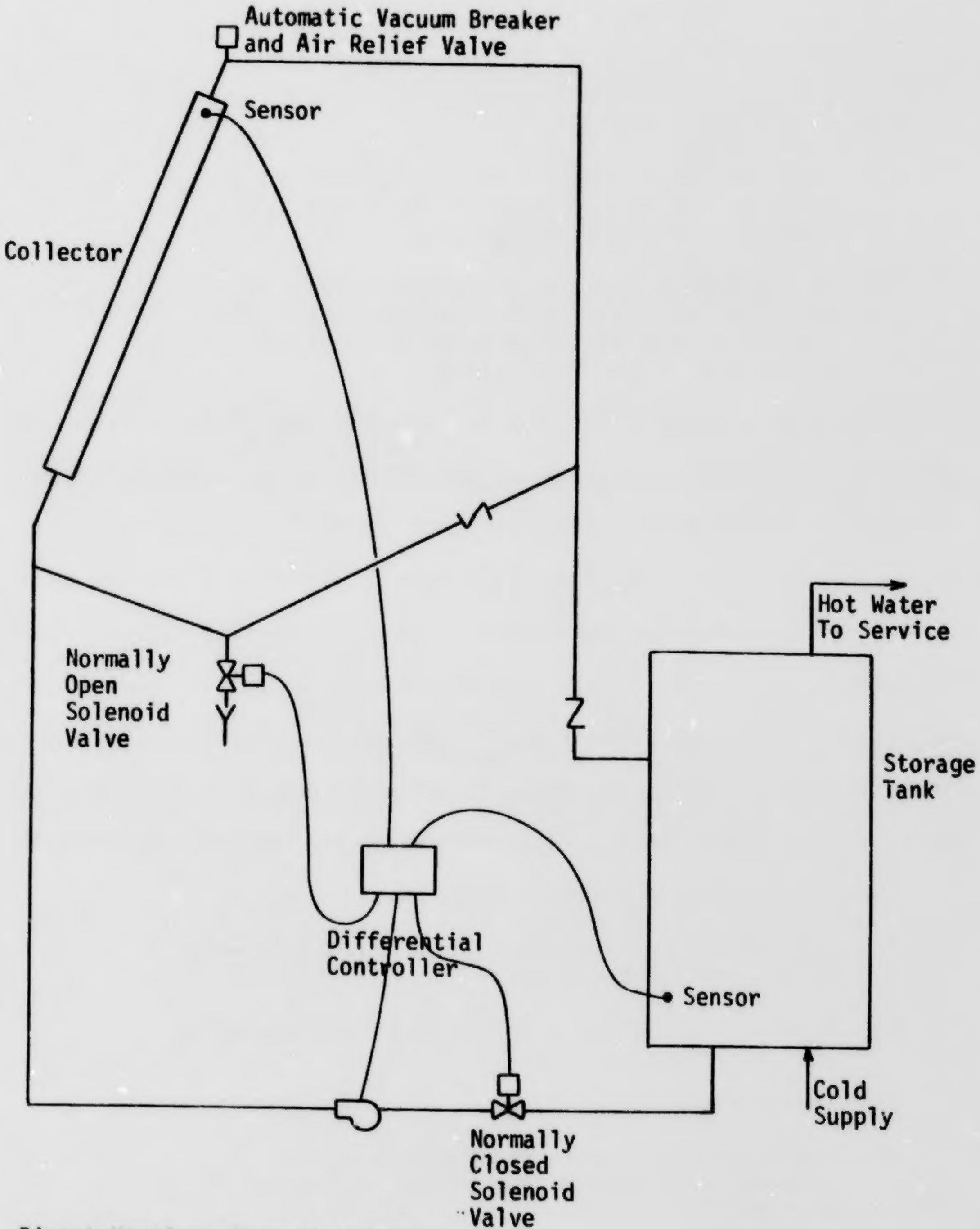
(1) frequent exposure of the wet piping and collector surfaces to air accelerates corrosion.

(2) all components and piping must be graded to permit complete drainage of outdoor components.


Partially Filled
Return Pipe
Collector
To Use
Thermal
Storage
Tank
Return

Gravity Return Drain Back
Solar Water Heater

FIGURE 4.1.3


Automatic Vacuum Breaker and Air Relief Valve
Sensor
Collector
Hot Water To Service
Normally Open Solenoid Valve
Storage Tank
Differential Controller
Sensor
Cold Supply
Normally Closed Solenoid Valve

Direct Heating, Pump Circulation Type of Solar Water Heater with Automatic Drain Down

FIGURE 4.1.4

(3) the circulation pump has to pump against a head equal to the highest point in the system, thus increasing the pumping requirements.

(4) if the thermal storage is not pressurized, a pressure system is required to integrate a domestic hot water system or other pressurized system into a drainback system.

A draindown system (Figure 4.1.4) overcomes some of the previous objections by operating a closed loop, pressurized solar energy system having controls to drain the system down when the differential temperature between the collectors and the storage tank is less than a present value or cold ambient conditions are sensed. Check and solenoid valves isolate the pressurized thermal storage tank from the solar loop and then drain the solar loop to a waste drain when signalled. Once satisfactory running conditions re-occur, automatic valves would revert to the operating positions, the system would re-fill with water and the pump would start.

The advantages of draindown systems are:

(1) This is a direct heating solar energy system with freeze protection.

(2) Pumping requirements are less than those of drainback systems.

(3) The system is pressurized.

(4) The frequency of exposure of piping to the atmosphere is reduced from that of drainback systems.

The disadvantages of a draindown system are:

(1) during transient solar conditions in cold weather, the system may cycle frequently, wasting water which has already been heated. This increases the operating costs (purchase of metered water) and decreases the system efficiency.

(2) as with the draindown system, piping must be properly graded to permit complete drainage.

(3) proper control operation is critical. Control failures may result in freeze damage to the system or may result in large volume water spills.

(4) automatic air vents and vacuum breakers may freeze after the system is drained and cooled, resulting in a water spill at the collector array.

For these reasons, drain down systems are most suitable in locations where the occurrence of freezing temperatures is rare. Most continuous use liquid solar energy systems designed for Canadian installations are Circulating Type, (Figure 4.1.5) commonly called dual fluid systems. These systems circulate the transfer medium, which is a non-freezing liquid, through the collectors where it absorbs solar energy before passing through a heat exchanger where its heat is transferred to a storage medium, usually water. The transfer media are not generally used for thermal storage because their heat storage capacities are less than that of water, and they usually are relatively expensive. The transfer medium is most commonly ethylene glycol and water or propylene glycol and water. Dual fluid systems overcome freezing problems at the expense of additional equipment, complexity and a slight decrease in system performance. Other disadvantages of dual fluid systems are related to the degradation of solar fluid and the necessity to prevent solar fluid from entering domestic water supplies.

Outdoor swimming pools present a very good opportunity for solar heating. The use period corresponds to that part of the year with high solar availability while frequently ambient temperatures will exceed pool

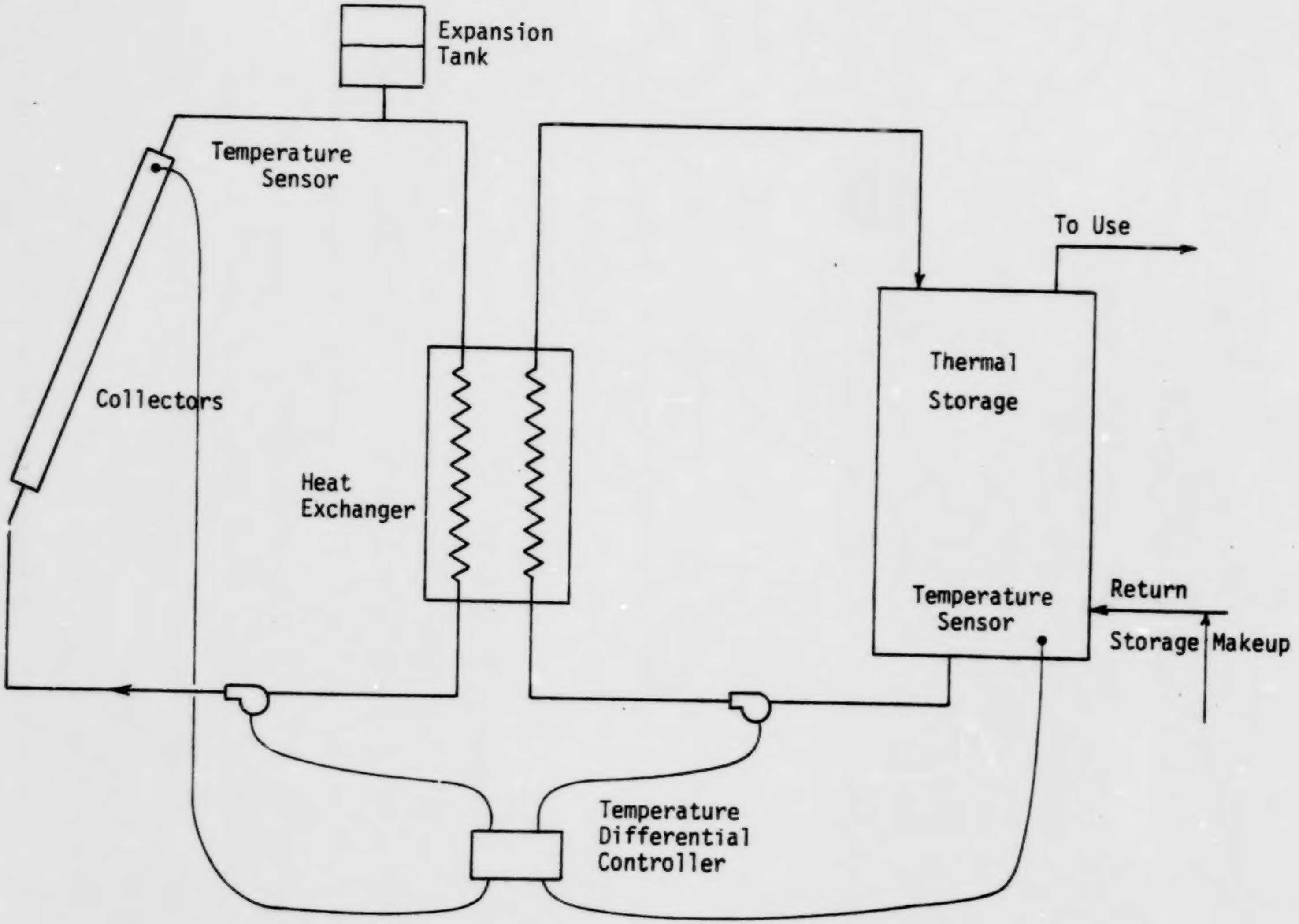

Expansion Tank
Temperature Sensor
Collectors
Heat Exchanger
To Use
Thermal Storage
Temperature Sensor
Return
Storage Makeup
Temperature Differential Controller

FIGURE 4.1.5
DUAL FLUID SYSTEM

temperatures. This means that the solar collector may gain energy from the ambient air as well as from solar radiation. Swimming pool systems are generally integrated into the pools filtering and auxilary heating circuits. The pool itself serves as thermal storage. The factors which make solar heating of swimming pools one of the most economical applications for solar energy include:

(1) The normal relationship between collector operating temperature and ambient air temperatures allows low cost, uninsulated collectors to achieve high operating efficiencies.

(2) Low operating pressures allow the use of low cost materials for collector absorber plates(such as extruded plastic sheets).

(3) Swimming pools have a large thermal mass which precludes the need for additional thermal storage.

(4) Summer only operation means the collector array is not very sensitive to slope or azimuth variation. This insensitivity frequently allows the engineer to design the system into existing site features, which will greatly reduce first costs.

(5) Summer only operation eliminates the need for freeze protection. Plastic collectors, which are frequently used in pool systems, can generally tolerate several freeze-thaw cycles without experiencing significant structural damage.

(6) Swimming pools have a relatively large heating load which occurs during that part of the year with a high availability of solar radiation.

(7) The simplicity of most swimming pool solar heating systems makes them technically and economically acceptable.

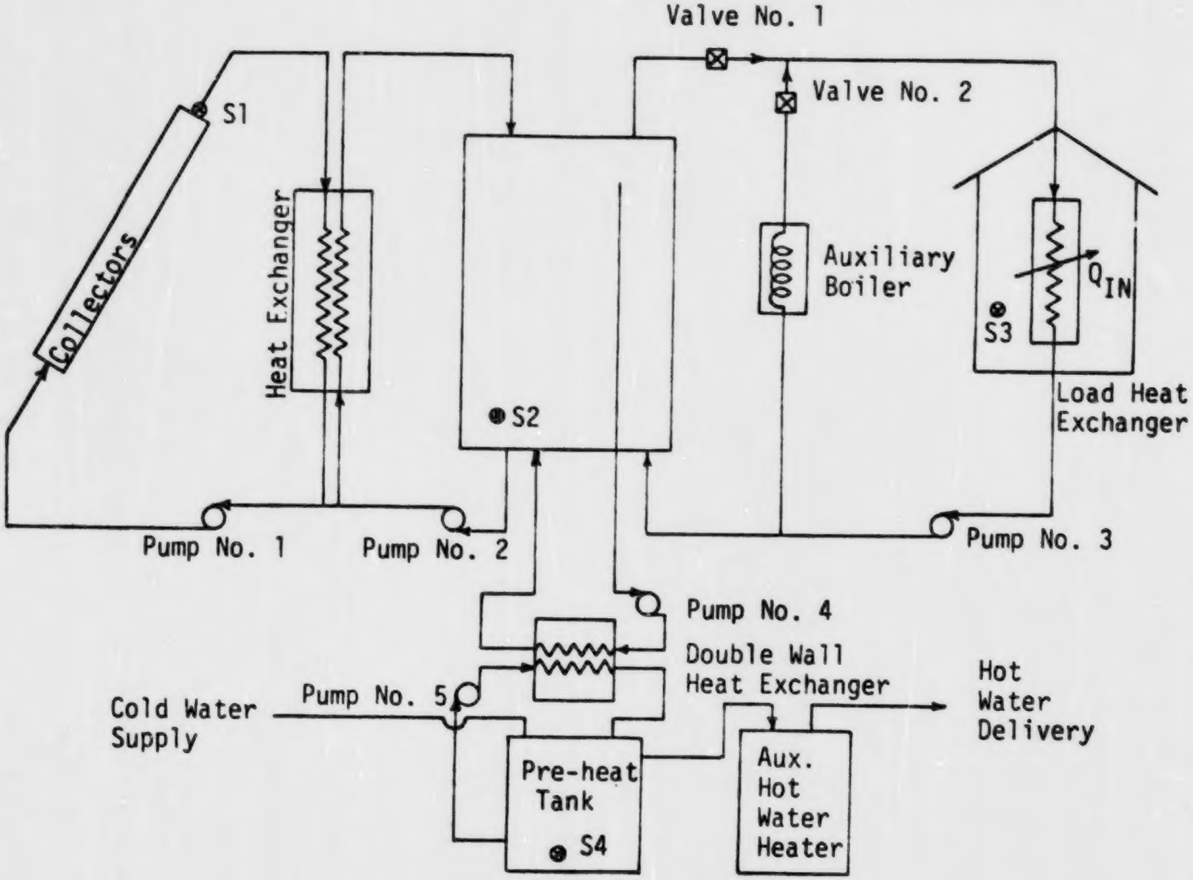
4.2 GENERAL DESIGN CONSIDERATIONS

4.2.1 Stored Heat To Use

The utilization of the collected energy determined its method of extraction from thermal storage. In a domestic water heating system, preheated water may be taken directly from thermal storage into the water system. In combined DHW - space heating sytems, the thermal storage may be contaminated by contact with heating system components, precluding direct use of storage water in the DHW system. A second heat exchanger is shown in Figure 4.2.1 is used to preheat the DHW in these systems.

4.2.2 High Temperature Protection

In solar DHW systems, there is a possibility that during times of low water use, DHW temperatures will exceed safe limits. The installation of a mixing valve downstream of the DHW heater will temper the water stream to the recommended temperature. DHW systems with recirculation pumps should incorporate a method of introducing water from thermal storage to the recirculation loop any time thermal storage temperatures exceed the DHW set point. This will reduce auxiliary heat requirements when DHW usage is low and excessive amounts of solar insolation are available. A further benefit is derived from the increased collector efficiency achieved as collector operating temperatures fall due to lower thermal storage temperatures.


Valve No. 1
Valve No. 2
S1
Collectors
Heat Exchanger
S2
Auxiliary Boiler
S3
Q_{IN}
Load Heat Exchanger
Pump No. 1
Pump No. 2
Pump No. 3
Pump No. 4
Double Wall Heat Exchanger
Cold Water Supply
Pump No. 5
Hot Water Delivery
Pre-heat Tank
S4
Aux. Hot Water Heater

Sensor Locations in a Typical Liquid-Heating System

FIGURE 4.2.1

If unsafe or excessive thermal storage temperatures can be reached during periods of high solar intensity and low heat utilization, a method of automatic heat disposal should be incorporated into the solar energy collection system. This mechanism should avoid excessive system pressures and temperatures and should minimize the occurrences of solar fluid blow-down.

4.2.3 Thermal Stratification

Temperature stratification in thermal storage should always be utilized to the advantage of the solar energy system. Water to be heated should be taken from the coolest sections of the thermal storage, that is the bottom. Heated water should be returned to the top of the thermal storage tank. Mixing in thermal storage should be minimized so that temperature gradients from the top to the bottom of the tank are maximized. Water for use should be taken from the top of the storage tank as this is the warmest segment. In a recirculating system, the water should be returned to the bottom of the thermal storage tank.

4.2.4 System Size

A general rule of thumb for a preliminary estimate of a solar energy collector system is that a space heating system should be sized to meet about 40% of the annual heating requirements, and water heating systems should be sized to meet about 60% of the annual water heating requirements. Final system sizing should be made on the basis of life cycle costing or other acceptable cost/benefit analysis methods.

In practice, there are often other factors affecting size decisions. These include the space available for mounting solar collectors, structural requirements, shading, orientation, load profiles and economic constraints. New construction can frequently be cheaply modified to accommodate optimum solar energy systems, while retrofit systems are restricted by existing conditions.

4.2.5 Collector Layout And Orientation

Although existing facilities may severely restrict the layout and orientation of the proposed solar energy system, these restrictions often are not as detrimental as might be expected. Variations of 10° to 15° from optimal slopes and azimuths have little impact on system performance. Depending on the energy usage profile, azimuths varying up to 60° from south may actually be beneficial to the system, by allowing reductions in thermal storage, or utilization of existing building features to reduce system costs, while not severely affecting solar contribution per dollar invested.

Collectors may be ground, wall or roof-mounted. Roof-mounting has the advantage of having fewer shading and vandalism problems. The designer should assess each application for the opportunity to incorporate solar collectors in an economical, maintenance free and aesthetically pleasing manner.

4.2.6 Roof Mounting

Placement of solar collectors on roofs may require a structural analysis of the roof. Support structures and associated plumbing will account for the majority of added weight. However, the major structural problem which may result from collector arrays on roofs is snow accumulation. Snow accumulation caused by solar collectors may be double the design snow loads in a given situation. For this reason, it is very important to perform a complete analysis to evaluate whether design loads will be exceeded.

Before a solar collector array is placed on a roof, all necessary roof repairs should be made. Re-roofing around collectors or a support structure, is considerably more costly and less effective than redoing an open roof.

Careful attention should be paid to tie down details. Collectors must be securely tied down to resist wind movement. Points of attachment and other roof penetrations must be waterproof and able to withstand movement due to wind, vibration and thermal expansion and contraction.

4.2.7 Pipe Sizing

Pipe sizing is a trade off between piping costs and pumping costs. A flow velocity of 1.5 m/s is the recommended maximum. Higher velocities may result in noisy pipes and excessive pumping energy requirements, as well as rapid wear. Some designer discretion is required in evaluating the cost/benefit ratios of various pipe sizes. High pumping requirements reduce the net energy gain by the system, which will increase the unit costs for the net energy gained. In well designed systems, electrical energy requirements of 5% of the solar heat delivered are possible while consumptions of 10% are common.

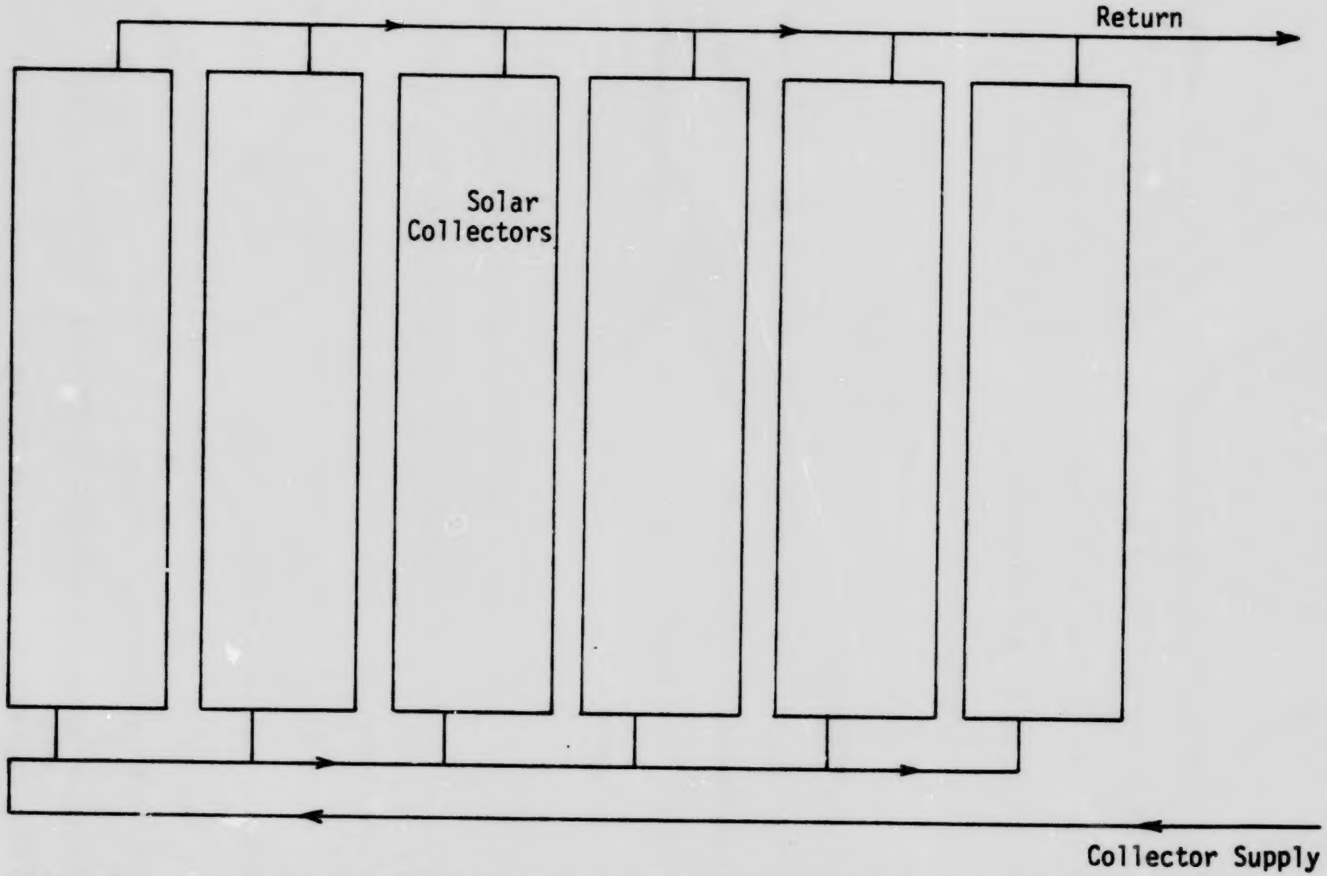
4.2.8 Piping Arrangements

Piping constitutes 10% to 40% of the costs in most solar energy systems, and therefore has the potential of significant cost savings. The length of pipe runs should be minimized to minimize cost, heat loss and piping requirements.

Piping should be graded to drain points to allow system evacuation for maintenance. If possible, the system should drain back to the tank which is used to fill the system. This will expedite the start-up after repairs are completed.

To induce balanced flow in all the collectors in an array, a piping arrangement called a reverse return is utilized. This arrangement (Figure 4.2.2) equalizes the pressure drop across the collectors and may eliminate the need for balancing valves.

Liquid collectors are generally plumbed for parallel flow. In some situations, it may prove desirable to use a series parallel arrangement in which a group of collectors is connected in series and then the groups in parallel. This will not affect the individual collectors' thermal performance provided the flow rate per unit area remains constant.


Return
Solar Collectors
Collector Supply

Piping Arrangement To Equallize Collector Fluid Flow Rates Through All Solar Panels

FIGURE 4.2.2

For example, if three collectors, whose recommended flow rate is 5 litres per minute, are connected in series, then the flow rate should be increased to 15 litres per minute. Note however, that series flow will increase pump head losses.

Occasionally, a flow rate or collector fluid is desired which is different from that used to establish the collector's thermal efficiency curve. To be strictly accurate, the collectors performance parameters, F_r $(\alpha)_e$ and F_r U_L, should be adjusted. However, if the correction factors cannot be easily obtained, they can generally be neglected without serious error provided the actual flow rate is at least 80% of the test flow rate.

4.2.9 Thermal Expansion

The solar fluid and the components in direct contact with the solar fluid experience large temperature fluctuations on both a daily and a seasonal basis. Components must have the capability to absorb the expansion and contraction associated with these temperature fluctuations. Operating temperature fluctuations must be accommodated through the use of expansion compensators. Collector absorber plates must be allowed to expand in the collector box while the connections between collectors take up this expansion. Outdoor solar fluid piping may experience up to 0.5% expansion during a year and about 0.3% in a day. This is much more than domestic water systems undergo, and the requirements for expansion compensators is proportionally greater.

The thermal expansion of solar fluid is dependent on the fluid. Between its freezing point and boiling point, water undergoes a 4% volume change. Glycol solutions may have volume fluctuations of 9% per 100°C, which can amount to over 15% over the annual operating temperature range.

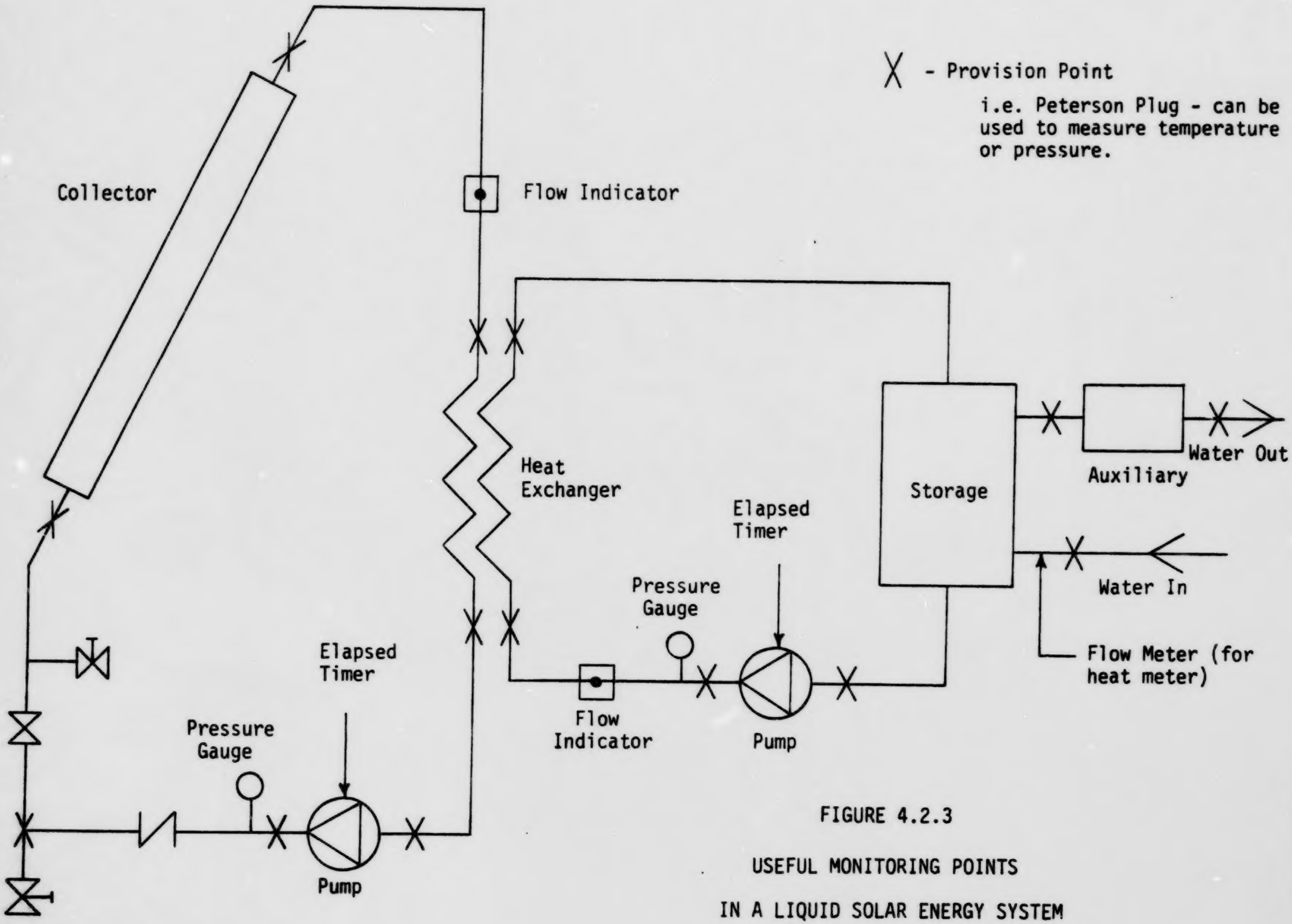
4.2.10 Galvanic Corrosion

An effort should be made to minimize the variety of metals in the solar energy system. Dielectric couplings should be used where dissimilar metals are joined. If severe corrosion problems are anticipated, as would occur in an aluminum-copper system, the installation of a sacrificial anode or an electrically induced galvanic protection system is recommended.

4.2.11 Plumbing Appurtences

Plumbing appurtences should be sufficient but not excessive. Strainers are necessary upstream of pumps. Valves should be located only where necessary for proper operation and maintenance procedures. By minimizing the number of valves, the possibility of inadvertently isolating a segment of the solar circuits is minimized.

Pressure gauges and thermometers located at key points in the system will permit easy system monitoring for trouble shooting and for performing system efficiency tests. Figure 4.2.3 shows the useful monitoring points in a liquid solar energy system. The use of Peterson plugs would permit a reduction in the cost and complexity of monitoring the system.



X - Provision Point
i.e. Peterson Plug - can be used to measure temperature or pressure.
Collector
Flow Indicator
Heat Exchanger
Storage
Auxiliary
Water Out
Water In
Flow Meter (for heat meter)
Elapsed Timer
Pressure Gauge
Flow Indicator
Pump
Elapsed Timer
Pressure Gauge
Pump

FIGURE 4.2.3

USEFUL MONITORING POINTS

IN A LIQUID SOLAR ENERGY SYSTEM

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CHAPTER 5

AIR SYSTEMS

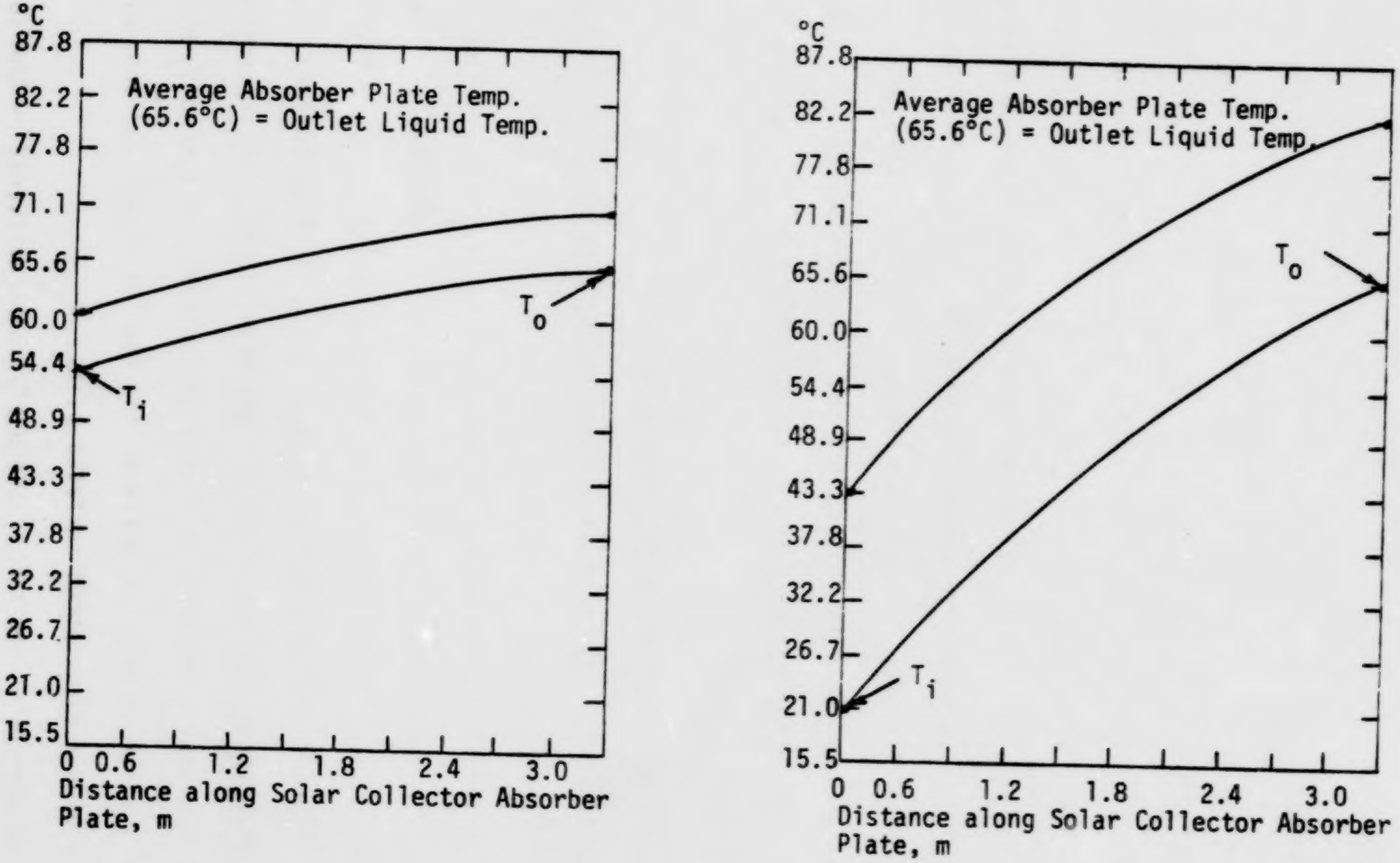
5.1 INTRODUCTION

Most of the flat plate solar collectors manufactured in Canada use a liquid as their working fluid. In the last few years though, production of air collectors has grown from zero to several hundred square metres per year. Solar energy system designers should therefore be familiar with the differences between the two types and should be conscious of what applications each is best suited to. This chapter discusses the performance and design of air collectors, and offers some guidelines for their proper application.

5.2 AIR COLLECTOR DESCRIPTION AND PERFORMANCE

The basis construction of most air and liquid collectors is the same, the major difference of course, being their choice of working fluids. A sketch of a typical air collector is shown in Figure 5.2.1. Notice that the air flows in direct contact with nearly the entire absorber surface and at a sufficient velocity for reasonably effective heat transfer. This is the major difference between the two types. Since fluid-to-surface heat transfer coefficients are much lower for air than liquids, it is necessary to increase the heat transfer area in air collectors. As a result, most designs use the entire absorber plate as the heat exchange surface, often with the assistance of extended surfaces (fins).

Under steady-state conditions, the useful heat delivered by the solar collector is equal to the radiant energy converted into heat at the absorber surface minus the heat losses from that surface. Expressed mathematically,

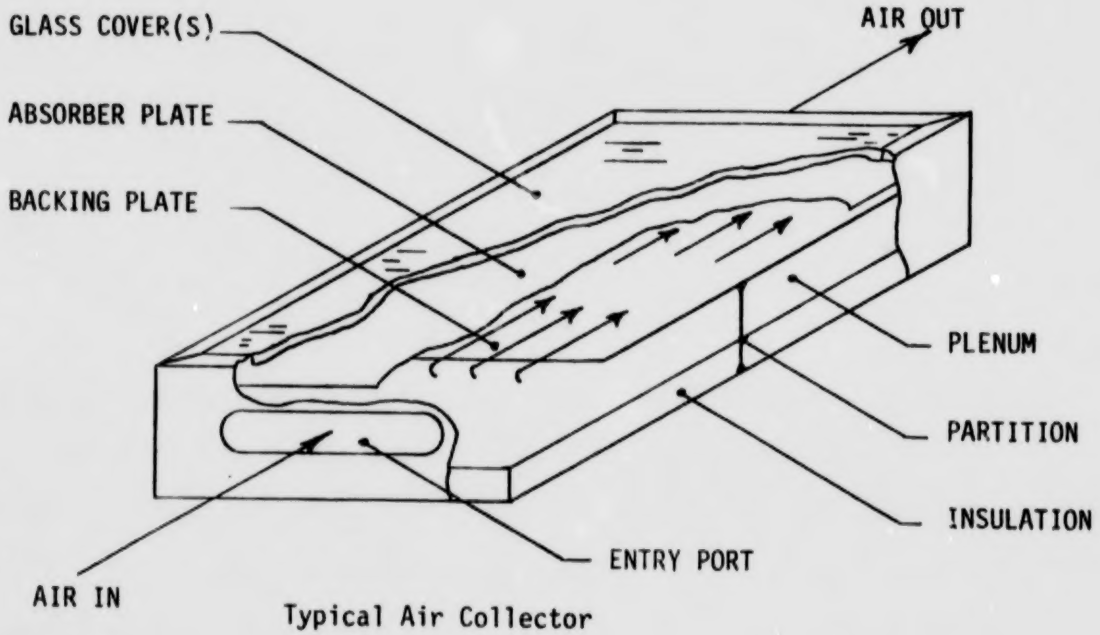


Air Heating Solar Collector

Air Heating Solar Collector

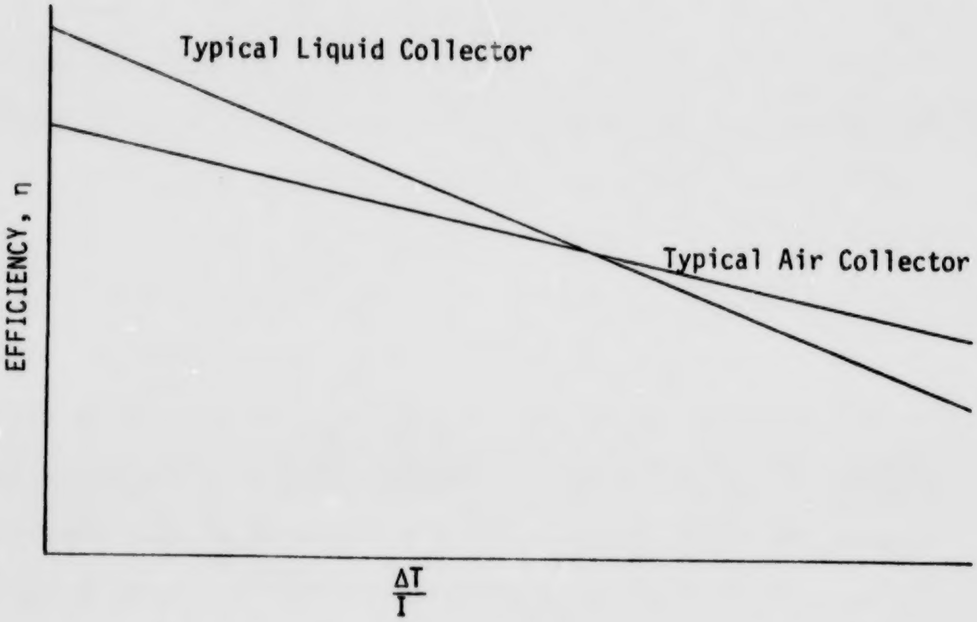
Comparison of Typical Temperatures in Liquid and Air Heating Solar Collectors

FIGURE 5.2.2


GLASS COVER(S)
ABSORBER PLATE
BACKING PLATE
AIR OUT
PLENUM
PARTITION
INSULATION
ENTRY PORT
AIR IN

Typical Air Collector

FIGURE 5.2.1



Comparison Of Typical Thermal Efficiencies
Of Air And Liquid Collectors

FIGURE 5.2.3

this is the familiar equation

$$Q_u = A_c F_r [(\tau\alpha)_e I - U_L (T_i - T_a)]$$

where Q_u = useful heat gain

and $F_r = \dfrac{\text{heat removed by collector}}{\text{heat removed if all fluid were at } T_i}$

= collector heat removal factor

Due to the low specific heat of air (approximately one-fourth that of water), F_r for most air collectors will range from 0.6 to 0.7. In contrast, F_r for liquid collectors is typically 0.9 or above. Since removal of equivalent amounts of heat from a collector (at equal mass flow rates) will cause air to experience a temperature rise four times that of water, the average operating temperature of an air collector will be higher than a liquid collector's at equal inlet temperatures and mass flow rates. This is shown in Figure 5.2.2. As a result, heat losses are greater for the air collector and F_r is lower. The smaller value of F_r means that the Y-intercept of the standard thermal efficiency curve, $F_r(\tau\alpha)_e$, will be lower, while the slope, F_rU_L, will be shallower than a typical liquid collector's is. This is shown on Figure 5.2.3.

This does not necessarily imply that liquid collectors produce greater amounts of energy than air collectors, but rather that the collector must be carefully matched to the anticipated load. For example, at high values of ΔT/I, an air collector may offer superior energy production. Another factor to consider is that heated air can be delivered at much lower temperatures and still be of value for space heating purposes. Thus, an air system may be able to operate at a lower ΔT/I than a liquid system and still do the same job.

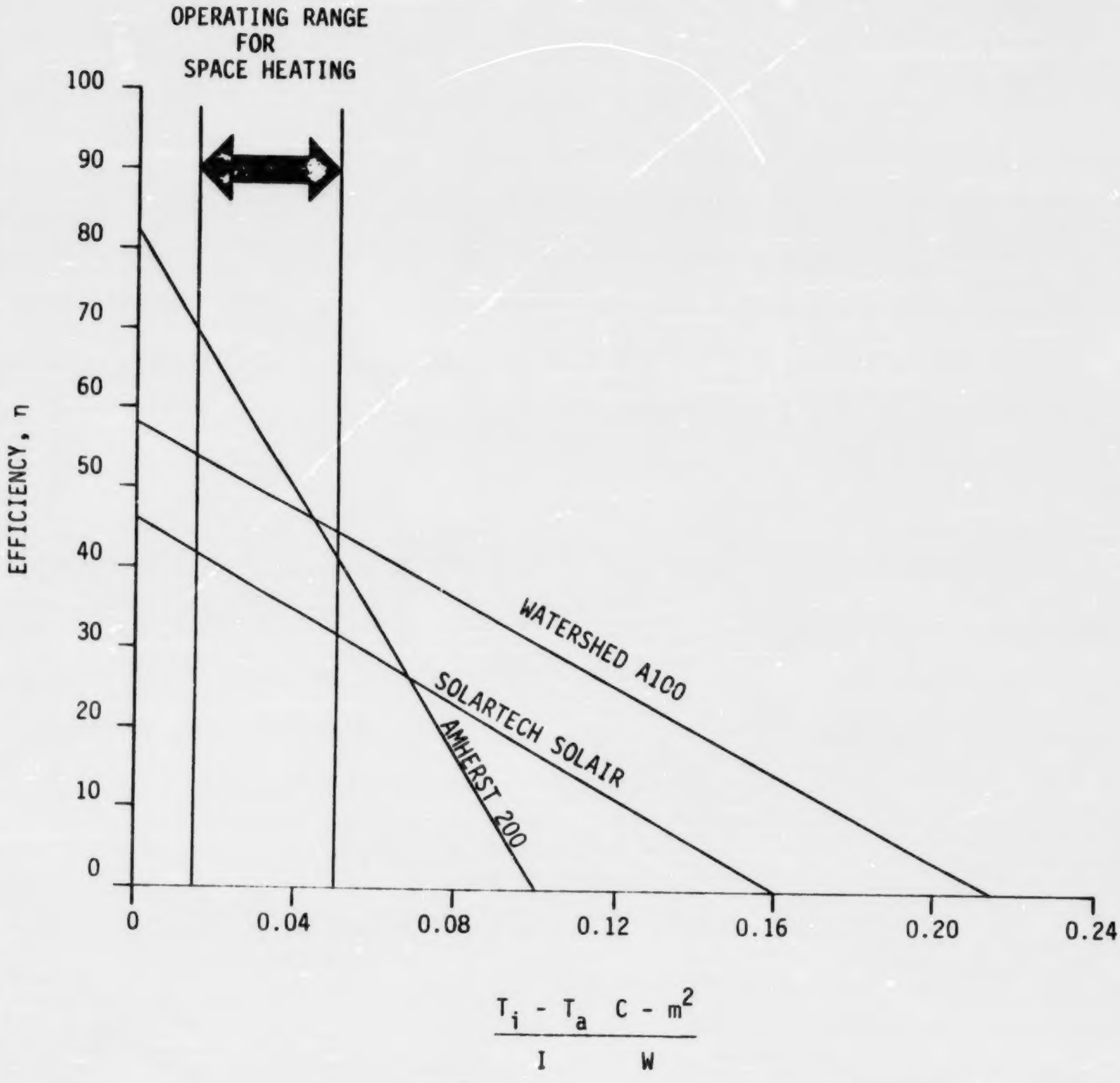

OPERATING RANGE
FOR
SPACE HEATING
100
90
80
70
60
50
40
30
20
10
0
EFFICIENCY, η
WATERSHED A100
SOLARTECH SOLAIR
AMHERST 200
0
0.04
0.08
0.12
0.16
0.20
0.24
$\frac{T_i - T_a}{I} \frac{C - m^2}{W}$

FIGURE 5.2.4

A comparison of efficiency curves for a number of air collectors is shown in Figure 5.2.4. Notice the wide variation between different designs. The Watershed collector, no longer in production, is shown as a dashed line because of the excessive scatter in the data from its efficiency test. The prototype overlapped plate was an experimental design intended to reduce the pressure drop through the collector; it has yet to enter production. The Solaron is an American collector which is produced in large quantities and probably represents the typical medium performance collector with acceptable reliability and durability. Both the Solartech and Amherst collectors are Canadian-made and currently in production. The Suntrap, also a Canadian design, is no longer in production.

Stagnation temperatures of most air collectors are approximately 100°C to 140°C. Their typical weight is 27 to 90 kg/collector or 9 to 40 kg/m^2.

Obviously, the final selection of a collector must consider other factors than just thermal performance. Cost, life, durability, availability, etc. all have to be investigated before the final selection can be made.

5.3 APPLICATIONS

The three most common applications of air systems are space heating, ventilation air preheating and crop drying.

5.3.1 Space Heating

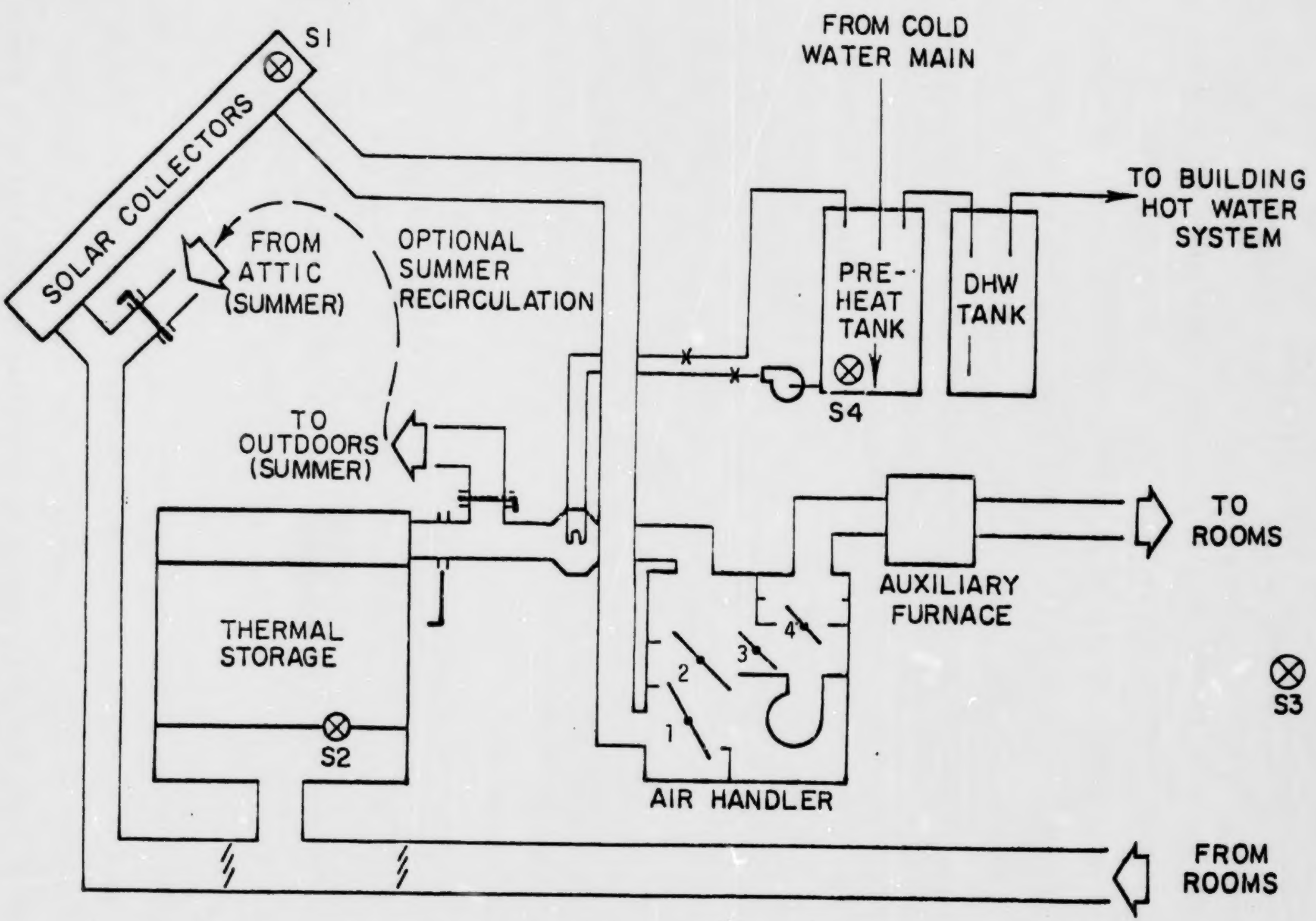
This has been the traditional use of most air collectors. Space heating applications generally require a full complement of auxiliary components, including storage, complex controls, and a separate air handling system. Inlet temperatures are typically 10°to 40°C above ambient and the operating efficiency ranges around 30% to 70%. Due to the comparative sophistication of the systems and their less than optimum operating regime (discussed in more detail below), space heating with air collectors can, at best, be described as only a fair application of solar energy in the Canadian environment.

A typical airspace heating system is shown in Figure 5.3.1. Its major components are the: collectors, pebble bed storage (shown in more detail in Figure 5.3.2), air handler, DHW preheat tank with coil and associated ducting. Note the position of the temperature sensors. The design shown uses a single blower and is intended primarily for residential applications. It has three major modes of operation which are controlled by the position of the motorized dampers in the air handler:

Mode of Operation	Damper Position			
	A	B	C	D
Collectors delivering heat to storage	open	closed	open	closed
Collectors delivering heat to rooms	open	closed	closed	open
Storage delivering heat to rooms	closed	open	closed	open

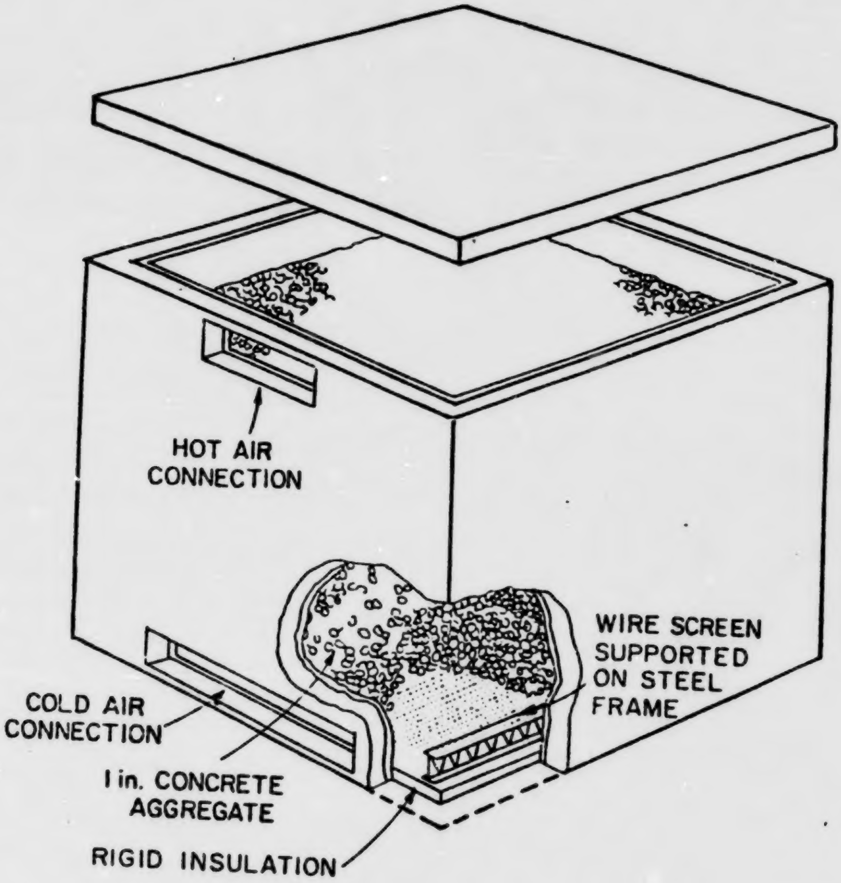
An optional summer recirculation mode for preheating the domestic hot water (and cooling the collectors) is also available using the three manual dampers shown.

Notice that the storage is always charged in one direction and discharged in the opposite direction. This allows significant thermal stratification to be developed in the pebble bed. Due to the poor heat transfer between adjacent pebbles, this stratification can be maintained for long periods of time.


FROM COLD WATER MAIN
S1
SOLAR COLLECTORS
FROM ATTIC (SUMMER)
OPTIONAL SUMMER RECIRCULATION
PRE-HEAT TANK
DHW TANK
TO BUILDING HOT WATER SYSTEM
S4
TO OUTDOORS (SUMMER)
TO ROOMS
THERMAL STORAGE
AUXILIARY FURNACE
1
2
3
4
S3
S2
AIR HANDLER
FROM ROOMS

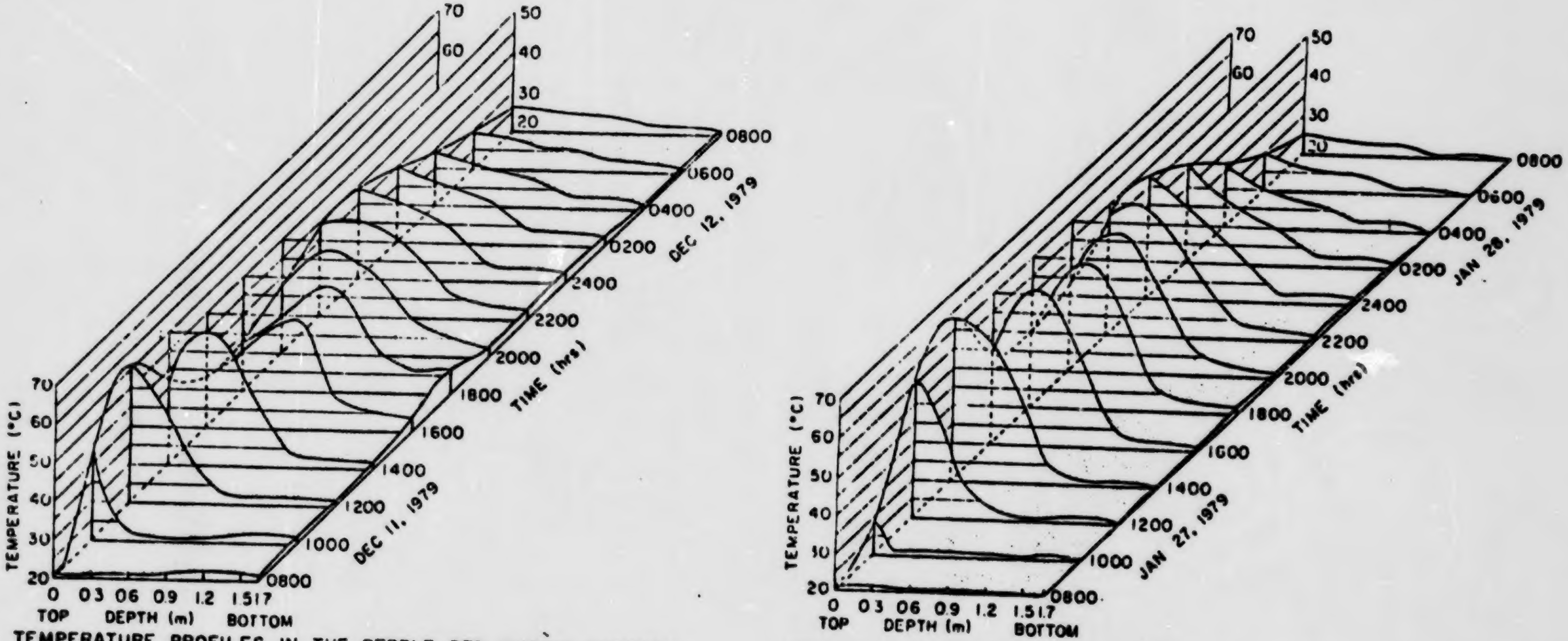
TYPICAL AIR SPACE HEATING SYSTEM

FIGURE 5.3.1


HOT AIR CONNECTION
WIRE SCREEN SUPPORTED ON STEEL FRAME
COLD AIR CONNECTION
I in. CONCRETE AGGREGATE
RIGID INSULATION

PEBBLE-BED HEAT STORAGE UNIT

FIGURE 5.3.2



TEMPERATURE PROFILES IN THE PEBBLE BED DURING CHARGE/DISCHARGE CYCLE AIR FLOW RATE 13.1 ℓ/(sec m²)

TEMPERATURE PROFILES IN THE PEBBLE BED DURING CHARGE/DISCHARGE CYCLE AIR FLOW RATE 9.6 ℓ/(sec·m²)

TYPICAL TEMPERATURE STRATIFICATION IN A PEBBLE BED

FIGURE 5.3.3

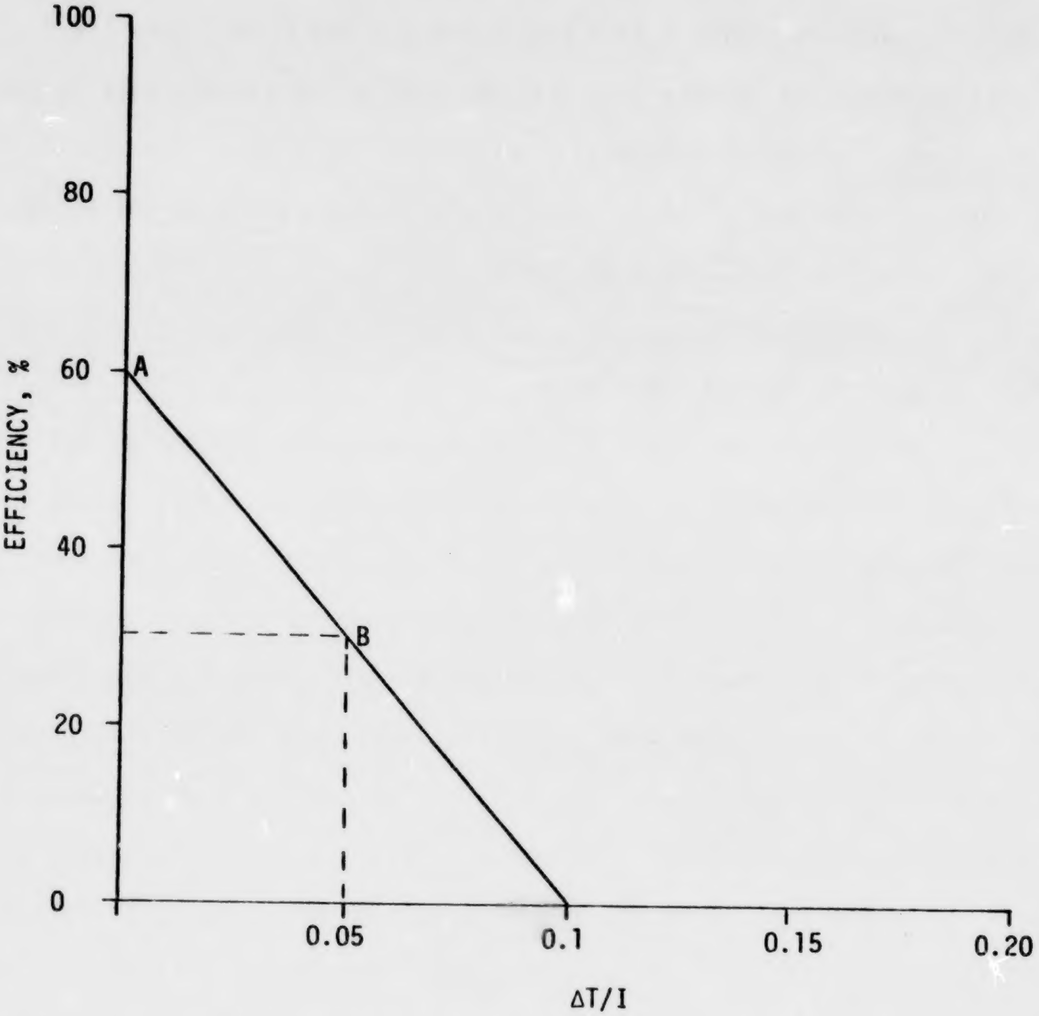
Typical temperature profiles in a pebble bed are shown in Figure 5.3.3.

It is important to note that pebbles used in the storage must be thoroughly washed to remove all dust.

Other, nonresidential, space heating applications which do not use any thermal storage have also been designed.

5.3.2 Ventilation Air Preheating

Many buildings, especially large commercial, industrial and agricultural structures, have significant ventilation air heating requirements. Because of their need to exhaust contaminated or moisture-laden air, fresh air has to be introduced into the structure and heated. Ventilation air preheating is a promising application for air collectors since they can operate at a high (and constant) efficiency, equal to the Y-intercept on the efficiency curve (since $\Delta T/I = 0$). Figure 5.3.4 demonstrates the thermal advantage of such an arrangement. System costs are generally quite low since no storage, automatic control system or mounting structure (if a suitable south-facing wall is available) is required. Since many applications are factory environments, trained maintenance personnel are normally available to service the system. Note that this may not necessarily apply to structures which have significant quantities of internal heat generation with comparatively small fresh air requirements, such as office buildings. In these cases, outdoor air is used not only for ventilation purposes, but also to prevent overheating of the building's core. In some applications with moderate internal heat generation, such as schools, a control system may be required which balances the (solar) preheated air with the incoming outdoor



The thermal efficiency curve shown is typical of many air collectors. Notice that by operating at point A (preheating), the efficiency is twice that available at point B (space heating), 60% versus 30%.

Importance of Proper Application For Efficient Collector Operation

FIGURE 5.3.4

air to provide the desired mixed air temperature. Since ventilation rates on many large commercial and industrial structures are often quite high, the designer should ensure that the collector's maximum flow rate is not exceeded.

The most desirable applications are those which have their entire requirement for outdoor air occurring during daylight hours, such as in a single shift factory. Then, the demand for preheating energy will occur simultaneously with the periods of available solar radiation. Ventilation air preheating is generally more economic on new construction than retrofits.

Of course, heat recovery may also be considered as a means to reduce the preheating load. Since heat recovery systems are easiest to install when the exhaust and outdoor air streams are adjacent to each other, solar preheating can best compete against heat recovery when the outdoor air inlets and exhaust air outlets are not physically in close proximity to each other.

On some installations, it may be necessary to filter the air prior to admitting it to the collectors.

5.3.3 Crop Drying

Many agricultural crops, such as grainery rice and tobacco, are harvested in a semi-moist condition. Drying is completed using low temperature, high volume air flows with the heating energy often being supplied by expensive fuels such as propane. Solar energy can be used to supply a portion or all of this load with the application being rather similar to ventilation air preheating. The major advantage of using air collectors, of course, is the high efficiency at which even simple, inexpensive models can operate.

The major disadvantages is that the drying system can only be used for a small fraction of the year, typically one to three months. Unfortunately the drying period also occurs during the fall and early winter months when available solar radiation is at a minimum.

5.4 DESIGN PROBLEMS

The design of any solar collector, whether air or liquid, requires careful consideration of many, often conflicting, requirements such as cost, durability, performance, weight, etc. These common design criteria are discussed in more detail in Chapter 6. However some of the unique problems which air collectors experience are described below.

1. Because of the low absorber-to-air heat transfer coefficients, it is necessary to significantly increase the heat transfer surface area and thus enlarge the total wetted area of the flow passage. This makes proper sealing of the flow channel(s) more difficult, with the result that air leakage into or out of the collector may seriously degrade performance. Measurements of air collectors have indicated that leakage may equal 5% to 10% (or more) of the total collector air flow rate. Proper sealing

of the flow passages is often difficult and expensive. As a result, a thorough check of the collectors (and the system), to find any leaks, should be carried out prior to final commissioning.

2. Degradation of the critical absorber plate surface due to condensation or abrasion may occur if the airflow is passing over the front of the plate.

3. Corrosion of the air flow channel(s) due to condensation may also result when the system is shut down and the collector is allowed to cool.

4. The use of air collectors generally necessitates the use of a good air filtration system, especially if the pebble bed storage system is used.

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CHAPTER 6

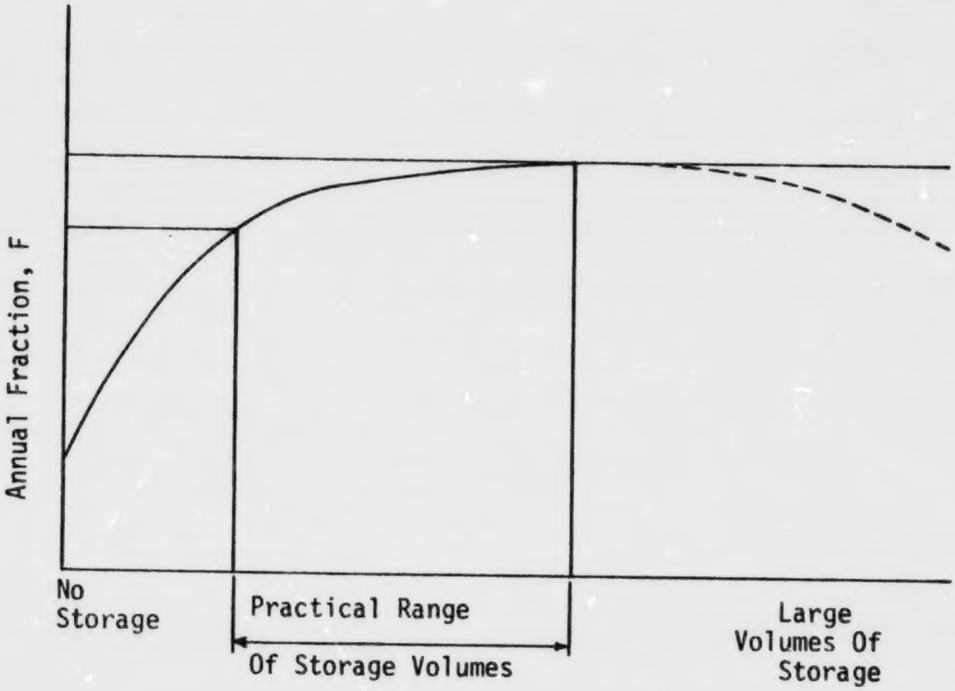
SYSTEM COMPONENTS

6.1 HEAT STORAGE

The purpose of heat storage is to capture and store excess solar energy when it is available and make it available for use when the incident solar energy cannot meet the heat load. The heat may be stored by raising the temperature of the storage medium (sensible heat), changing the phase of the storage medium (latent heat) or by utilizing certain chemical reactions. The heat is recovered by reversing the storage operation. Long term thermal storage, although of interest, is still experimental and is not covered in depth here.

6.1.1 Sizing Thermal Storage

The effect of storage size on solar heat contribution is shown in Figure 6.1.1. With no storage, only energy collected during times of energy demand can be utilized. Storage permits a delay between the time of collection and utilization of the solar energy. At first the increase in solar fraction rises rapidly as storage size increases, but it increases at a diminishing rate. As storage is increased beyond a certain size in a DHW system, solar fraction begins to decrease. This is due to a decrease in storage temperature which results in a lower availability of useful energy. Also skin losses from the storage tank increase with surface area. Thus, excessive storage volumes actually degrade the performance of DHW systems.



Effect of Storage Size on Solar Heat Contribution to a DHW Load

FIGURE 6.1.1

The exact amount of storage required is dependent on the heating load profiles. In a residential hot water system where most of the load occurs in the morning and the early evening, when incident radiation is usually low, storage is a critical component in the solar energy system, and should be sized to hold about one day's hot water requirements.

A processing plant which has a continuous hot water draw and a modestly sized solar energy system would not require any solar storage because the energy would be utilized as it was collected.

For most applications, the practical range of thermal storage is 200 to 300 kJ/°C m^2 of collector area (50 to 75 litres of water storage per square meter of collector). A rule of thumb is to size storage to meet a one day heat load or one day's incident radiation.

Water is the most common storage medium used in liquid systems because of its price, good heat storage capabilities, and its consistency of viscocity and thermal properties in the relevant range.

The solar fluids used in the solar collector loop can be used for thermal storage, but these are usually expensive and have less favourable thermal, chemical and mechanical properties than water. Tables 6.1.1 and 6.1.2 give properties and comparisons of some typical solar fluids while Table 6.1.3 lists some by trade name.

Fluid storage tanks can be made of a variety of materials and in several configurations. The design will depend on the location of the storage, and the storage temperature and pressure. In all cases, storage should be designed to promote thermal stratification, which can be bene-

Table 6.1

EXAMPLES OF TYPICAL HEAT TRANSFER LIQUID PROPERTIES

Property	Water	Glycols		Silicone Fluid	Hydrocarbons		Glycerine ("Glycerol") 60%/40% Glycerol/water
		50% Ethylene Glycol/Water	50% Propylene Glycol/Water		Aromatics	Paraffinic Oil	
Freezing point, (°C)	0	-36	-33	-50	-62 to -9	-9	-35
Boiling point, (°C)(at atm. pressure)	100	110 (197 for 100% glycol)	106 (180 for 100% glycol)	None	181-337	371	110
Fluid Stability	Requires pH or inhibitor	Requires pH or inhibitor monitoring	Requires pH or inhibitor monitoring	Good	Good to Fair	Good to Fair	Good
Flash Point (°C)	None	None (111 for 100% glycol)	None (99 for 100% glycol)	232-315	56-207	149-235	None (177 for 99% glycerol)
Specific Heat 23°C (KJ/KG/°C)	4.18	3.35	3.56	1.42-2.01	1.51-2.09	1.80-2.64	3.35
Thermal Conductivity at 38°C kW/cm °C	6.28×10^{-3} @ 0°C	3.98×10^{-3}	3.89×10^{-3}	1.44×10^{-3}	1.21×10^{-3}	1.21×10^{-3}	4.67×10^{-3}
Kinematic Viscosity $\times10^{5}$ m^2/s at 25°C	0.083	0.316	0.464	1.86-4.64	0.744-4.64	0.093-5.58	0.651
Toxicity	Depends on inhibitor used	Moderate to high depends on inhibitor used	Slight depends on inhibitor used	No unequivocal toxic effects are recognized	Moderate	Low	Slight
Lifetime Estimated	Depends on inhibitor used	Less than 2 years	Less than 2 years	More than 5 years	Ranges 1-20 years	On order of 10 years	More than 5 years

TABLE 6.1.2

GENERAL COMPARISON OF TYPICAL HEAT EXCHANGER FLUIDS

FLUID	ADVANTAGES	DISADVANTAGES
Water, untreated	1. nontoxic 2. environmentally safe 3. inexpensive 4. high thermal efficiency	1. freezes at 32°F 2. supports galvanic corrosion 3. boils at 212°F 4. promotes scale formation 5. may change composition in closed system
Water/glycol mixtures	1. will not freeze down to -35°F 2. inhibitor technology for ethylene glycol well-suited to offering protection in a multimetal system 3. boils at temperatures higher than water	1. fluid must be replaced frequently or maintained properly (by monitoring ph) 2. problem of disposal of large amounts 3. decomposes around 280 - 300°F forming sludge and organic acids 4. toxic (ethylene glycol) 5. can degrade building material (roofs)
Hydrocarbons	1. low cost 2. nonvolatile 3. less toxic than ethylene glycol	1. poor oxidation stability 2. poor thermal stability at high temperatures resulting in sludge and acid formation 3. high viscosity at low temperatures 4. environmental effects similar to motor oil 5. typical closed-cup flashpoints run 300 - 320°F, fluids with higher flashpoints have a higher viscosity 6. damage to building from leakage may be irreparable due to odor and low volatility 7. low heat conductivity
Silicones	1. do not freeze 2. do not boil 3. do not corrode common metals 4. long life 5. high flashpoint 6. low toxicity, if any	1. low heat capacity 2. high viscosity 3. expensive 4. not biodegradable 5. difficult to seal 6. strong tendency to leak 7. may require more expensive pump (may also be true for hydrocarbons)

Based on mfr. claims

TABLE 6.1.3

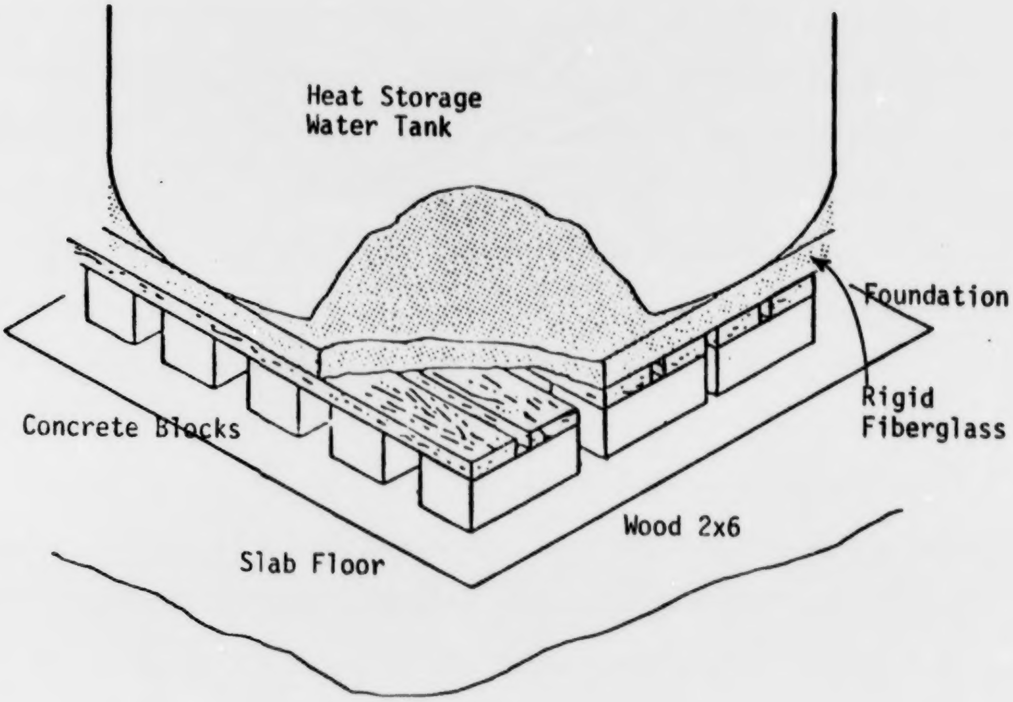
SOLAR HEAT TRANSFER FLUIDS

TRADE NAME	MANUFACTURER	COMPOSITION
•UCAR-17	Union Carbide	Ethylene glycol
•DOWTHERM SR-1	Dow Chemical	Ethylene glycol inhibitor < 5% and red dye < 1%
UCON-500	Union Carbide	Polyglycols - may contain phenyl β naphthyl amine < 10%
•UCON-30	Union Carbide	Ethylene oxide
•DOWFROST	Dow Chemical	Propylene glycol - contains inhibitor
•UCAR-35	Union Carbide	Propylene glycol
MOBILTHERM 603	Mobil Oil	Paraffinic, neutral oil
MOBILTHERM LIGHT	Mobil Oil	Paraffinic, high aromatic oil
SUNTEMP 1	Resource Technology Corp.	Paraffinic, low aromatic oil contains red dye (< 1%)
CALORIA HT-43	Exxon	Paraffinic, low (14%) aromatic oil
THERMIA 33	Shell	Paraffinic oil - aromatic content 20 - 25%
PROCESS OIL-3029	Exxon	Naphthenic mineral oil aromatic content < 10%
•DOW CORNING 200	Dow Corning	Silicone oil
SF 96	General Electric	Polydimethylsiloxan
•THERMINOL 66	Monsanto	Terphenyl mixture
•DOWTHERM A	Dow Corning	Diphenyl and diphenyl oxide mixture
SOLARGARD G	Daystar	Glycerol (USP) 60%/water 40%

ficial to system efficiency.

The most desirable location for a storage tank is indoors. Outside and underground storage tanks are subject to corrosion and deterioration of insulation while heat losses from indoor tanks can meet part of a building's heat requirements. For high temperature or high pressure storage, steel tanks are recommended. Commercial grade hot water pressure storage tanks are available in a variety of configurations and sizes but are quite expensive. Light gauge metal tanks, fiberglass reinforced plastics (F.R.P.) tanks and wood or masonary tanks lined with plastic membranes are moderately priced alternatives for unpressurized thermal storage. It may be convenient to utilize foundation walls for one or more of the tank sides. Tanks that are constructed in place have the advantage that their dimensions are not restricted by the size of doorways and passages. Square or rectangular storage tanks can be easily and neatly insulated. Storage tanks should be insulated from direct contact with walls and floors. Tank linings or waterproof membranes should be flexible and strong enough to resist damage if the tank walls or footings shift or crack. The tank and its membrane should be accessible for repairs or modifications, should the need arise.

Insulation levels of RSI5 (R30) are recommended for outdoor thermal storage and RSI2 (R 10-12) is recommended for indoor thermal storage. Flat bottomed tanks should sit on rigid insulation to reduce bottom losses. (Figure 6.1.2) Saddle supports under cylindrical tanks are quite difficult to insulate, but the effort is worthwhile. Care should be taken to


Heat Storage
Water Tank
Foundation
Rigid
Fiberglass
Concrete Blocks
Wood 2x6
Slab Floor

Bottom Insulation and Support Scheme for Flat-Bottom Water Storage Tanks

FIGURE 6.1.2

ensure that insulation on the floor does not stand in water, as wet insulation may lose insulating properties.

Underground tanks must be insulated with a material that does not absorb moisture and does not degrade. Tanks should be located below the frost line and above the water table, which precludes their use in most Canadian localities. If possible, underground installations should be avoided.

A method to drain or pump out tanks should be included in the storage system design. Dielectric connections should be made where piping and tanks are of dissimilar metals.

Provision must be made for expansion and contraction of the liquid storage. In open storage tanks this involves leaving sufficient freeboard to accommodate any thermal expansion, or providing an overflow drain. In closed pressure systems, expansion tanks or pressure relief valves are necessary to avoid damage caused by excessive pressures. (Figure 6.1.3)

Provisions must also be made to permit disposal of excess heat. In open systems, permitting the water to boil is one solution. This results in a buildup of scale in the storage tank and necessitates having a make-up water system. The high temperature can also be hard on the collectors.

It is important in pressurized systems to incorporate temperature relief valves in the storage system. If the "blow off" temperature is much above 100°C, the water will flash to steam as it leaves the tank. This is a hazard requiring special consideration. By blowing off hot storage water and introducing cold make-up water, safe storage temperatures

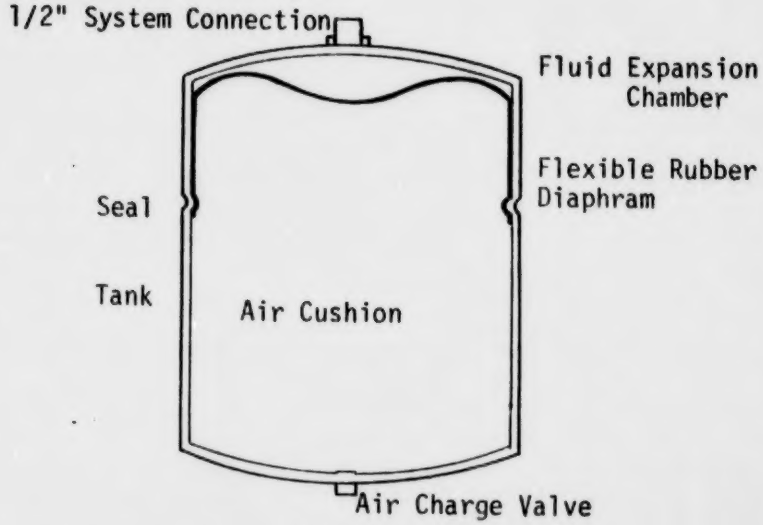

1/2" System Connection
Fluid Expansion Chamber
Flexible Rubber Diaphram
Seal
Tank
Air Cushion
Air Charge Valve

FIGURE 6.1.3

can be maintained.

Other methods that can be used to avoid excessive storage temperatures include:

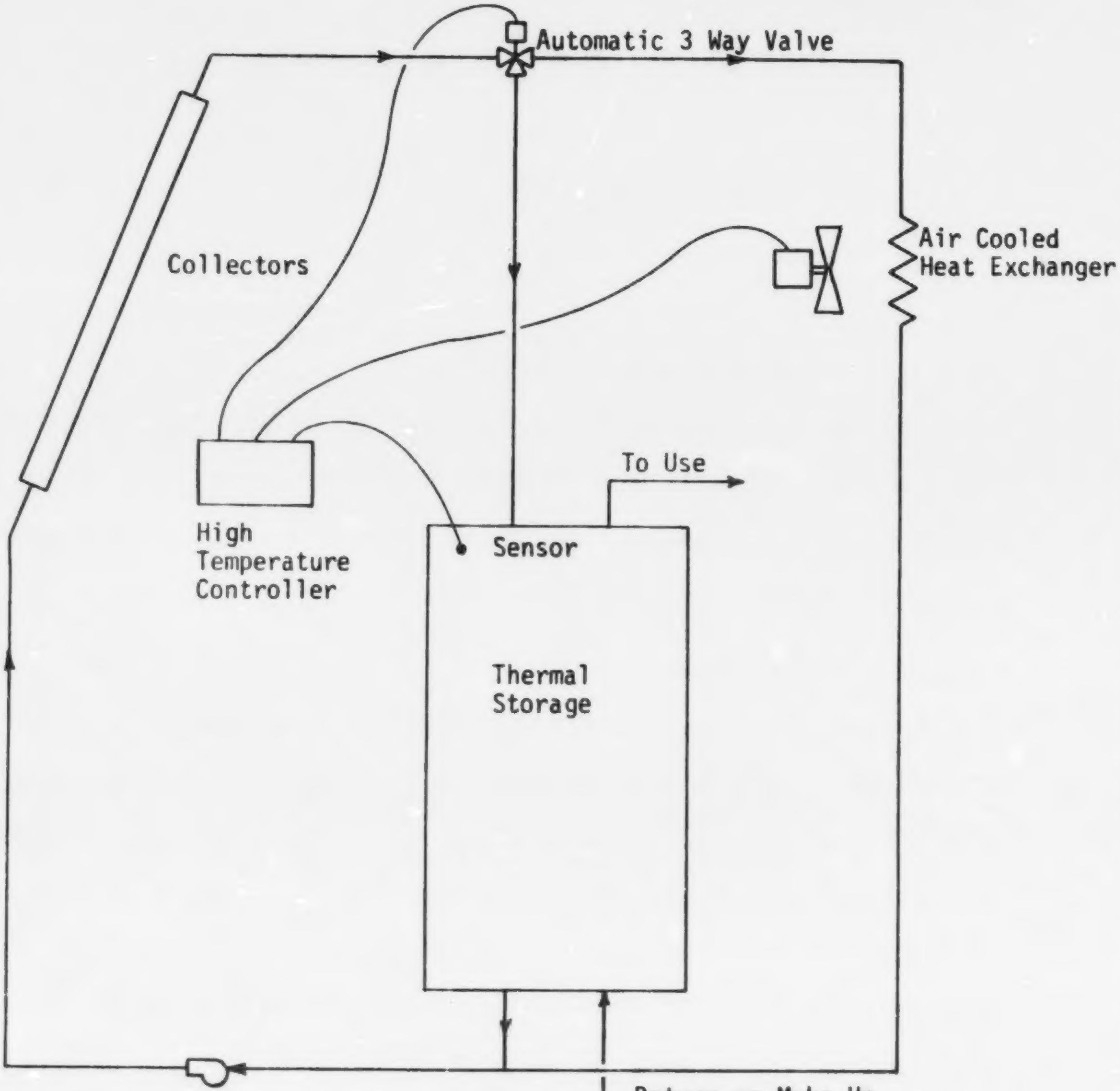
(1) having a fan coil arrangement which cools the fluid with outside air (Figure 7.1.4).

(2) having a high temperature shut-off on the collector pump, which would result in boiling in the collectors instead of in storage. The collectors must be able to stand stagnation temperatures and be equipped to allow the steam to escape.

When boiling in the collector or in storage is expected, water treatment with softeners and corrosion inhibitors is recommended. Glycol solutions are not recommended for use above 150°C for at these temperatures, chemical breakdown into acids is accelerated.

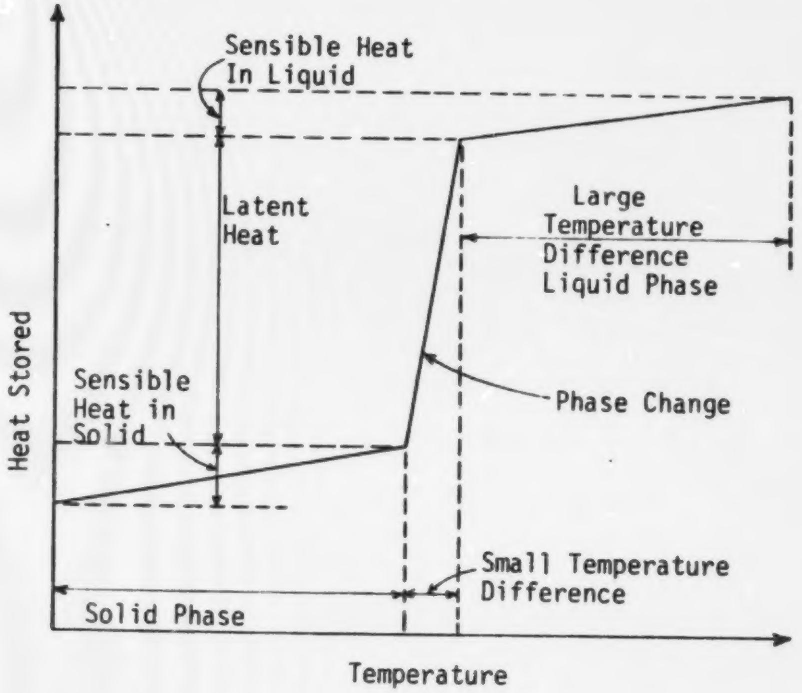
6.2 PHASE CHANGE MATERIALS (PCM's)

As their name suggests, PCM's utilize the latent heat of fusion for thermal energy storage with the heat being stored by liquifying the material (Figure 6.2.1). The advantages of PCM's are the reduced storage volume required to store a given quantity of heat, and the near constant temperature of the solar storage. Relatively low storage temperatures are conducive to high collector efficiency and constant temperatures simplify system design. In passive systems, low temperature PCM's are used to prevent the building space from overheating during the day and from excessive cooling at night. In this particular type of application, where only limited temperature swings are permissible, the volume of water or masonry necessary is very great - in the order of 10 times that required when PCM's are used.


Automatic 3 Way Valve
Collectors
Air Cooled
Heat Exchanger
To Use
High
Temperature
Controller
Sensor
Thermal
Storage
Return or Make-Up

Automatic Air Cooled Heat Dumping Circuit

FIGURE 6.1.4


Sensible Heat
In Liquid
Latent
Heat
Large
Temperature
Difference
Liquid Phase
Heat Stored
Sensible
Heat in
Solid
Phase Change
Small Temperature
Difference
Solid Phase
Temperature

Heat Storage In Phase Change Materials

FIGURE 6.2.1

Desirable characteristics for PCM's include:

(1) appropriate melting point

(2) little or no subcooling required to initiate solidification

(3) high heat of transition

(4) chemical stability (i.e. no loss of storage capacity over time)

(5) low cost

(6) nonflammability and nontoxicity

(7) small volume change on transition

(8) high thermal conductivity

(9) low vapour pressure

Two general classes of PCM's of current interest for solar use are organic compounds such as paraffin, and salt hydrates such as Sodium Sulphate Decahydrate, commonly called Glauber Salts. Organic PCM's tend to be more stable than salt hydrates but they have broad melting ranges which spread the heat of transition out over large temperature ranges. Other limitations of organic PCM's include low thermal conductivity, high flammability and high volume changes during melting. In salt hydrates, hydration and dehydration occur rather than simple freezing and melting. Salt hydrates are classified according to their phase change properties as (1) congruent (the solid and liquid phase composition have the same ratio of salt to bound water (simple freeze-melt)), (2) semi-congruent (the material has two or more hydrate forms with different melting points) and (3) incongruent (the solid and liquid phases have different ratios of salt to bound water which results in an insoluble anhydrous salt precipitate which, if allowed to settle out, results in reduced thermal storage capacities). Congruent salts with desirable characteristics are too expensive for most solar storage applications. The less expensive incongruent and semi-congruent salts have been the object of most research with most efforts directed at overcoming their problems.

The problems of getting the liquid solution to crystalize without subcooling has been overcome by using nucleating agents and by the use of a cool surface or "cold finger" in the PCM which promotes nucleation.

The settling out of precipitates in incongruent salts can be reduced in one of three ways. The salt hydrates in storage may be mechanically mixed, the crystals may be suspended in peat moss by the addition of thickeners or the material may be packaged in very thin horizontal layers (which reduces settling distances).

Containers housing PCM's which permit thermal expansion, are corrosion resistant, inhibit settling and encourage heat transfer and nucleation, can cost more than the PCM's they contain. To date, cost is the major drawback to the widespread use of PCM's in thermal energy storage systems.

TABLE 6.2.1

CHARACTERISTICS OF PHASE CHANGE MATERIALS

Name	Melting point ° C	Heat of Fusion	
		KJ/Kg	KJ/ℓ
Hydrated Inorganic Salts			
Sodium chromate	19.9	181.8	275
Manganese nitrate	26.3	140.1	245
Ortho Phosphoric acid	29.5	142.8	261
Lithium nitrate	29.9	294.5	677
Calcium chloride	30.2	169.7	282
Glauber Salt	32.4	237.2	346
Disodium phosphate	34.6	280.6	430
Manganous nitrate	35.4	116.9	232
Zinc nitrate	36.2	130.1	267
Calcium nitrate	42.6	141.7	257
Thiosulfate (Hypo)	48.6	94.4	175
Nickelous nitrate	56.7	152.1	310
Cobaltous nitrate	57.1	126.1	309
Cadmium nitrate	59.5	106.0	259
Sulfur trioxide	62.2	317.7	623
Magnesium nitrate	83.4	159.5	349
Hydrazine hydrochloride	92.7	221.9	287
Magnesium chloride	117.8	168.6	335
Anhydrous Inorganic Salts			
Arsenic tribromide	31.9	37.1	
Meta Phosphoric Acid	42.6	107.1	
Phosphoric acid	70	156.3	
Antimony trichloride	73.3	55.7	
Antimony tribromide	96.4	38.5	
Aluminum bromide	97.9	42.2	
Ammonium acid sulfate	143.9	124.1	
Ammonium nitrate	169.4	63.5	
Potassium thiocyanate	176.7	111.3	
Waxes and Organic Solids			
Anthracine	96.1	104.8	130
Anthraquinone	285	157.2	224
Naphthaline	80	150.5	172
Naphthol	95	162.3	197
Bees wax	61.7	176.7	167
Stearic acid (tallow)	76.1	198	167
Amorphous paraffin wax	74.4	229.6	182

6.3 CONTROLS

The controls in a solar energy system must operate the system so that it automatically collects and stores solar energy when it is available, and co-ordinates the application of this stored energy with the requirements of the energy user. Energy collection should be maximized, while electrical energy consumption should be minimized. In the event of a controls failure, the system should be designed to "fail safe". All this should be done simply, logically, dependably and cheaply.

The three basic components (Figure 6.3.1) of a control system are sensors, comparators and output devices. The sensors (usually temperature sensors) detect or measure variables at pertinent locations in the system. The comparators compare the conditions indicated by the sensors to some preset value or to the value measured by another sensor. This comparison is used to activate the output devices according to a prearranged plan. The output devices are mechanical components such as automatic valves, motor starters on fans or pumps and variable speed motor controllers.

The simplest and cheapest controllers are on-off units. Modulating controllers are more complex and costly, but lend themselves to more efficient system operation. However, the increased effectiveness is rarely worth the increased cost.

Controls are easiest understood if they are considered in sections. Usually a control system can be broken into a number of logical and separate segments. For example, the system shown in Figure 6.3.2 can be separated into three sections:

(1) collector to storage loop control.
(2) storage to preheat loop control.
(3) storage to space heating loop control.

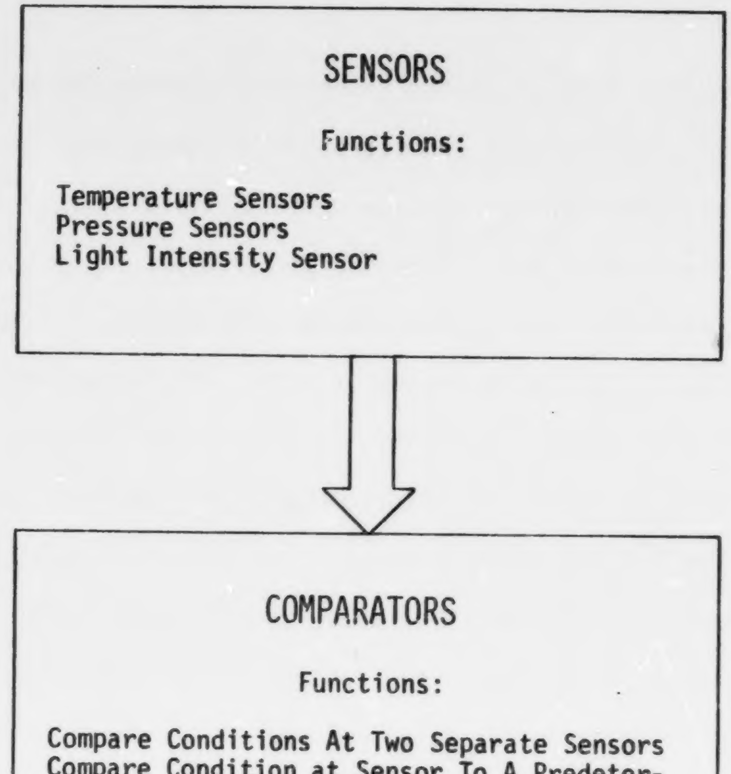
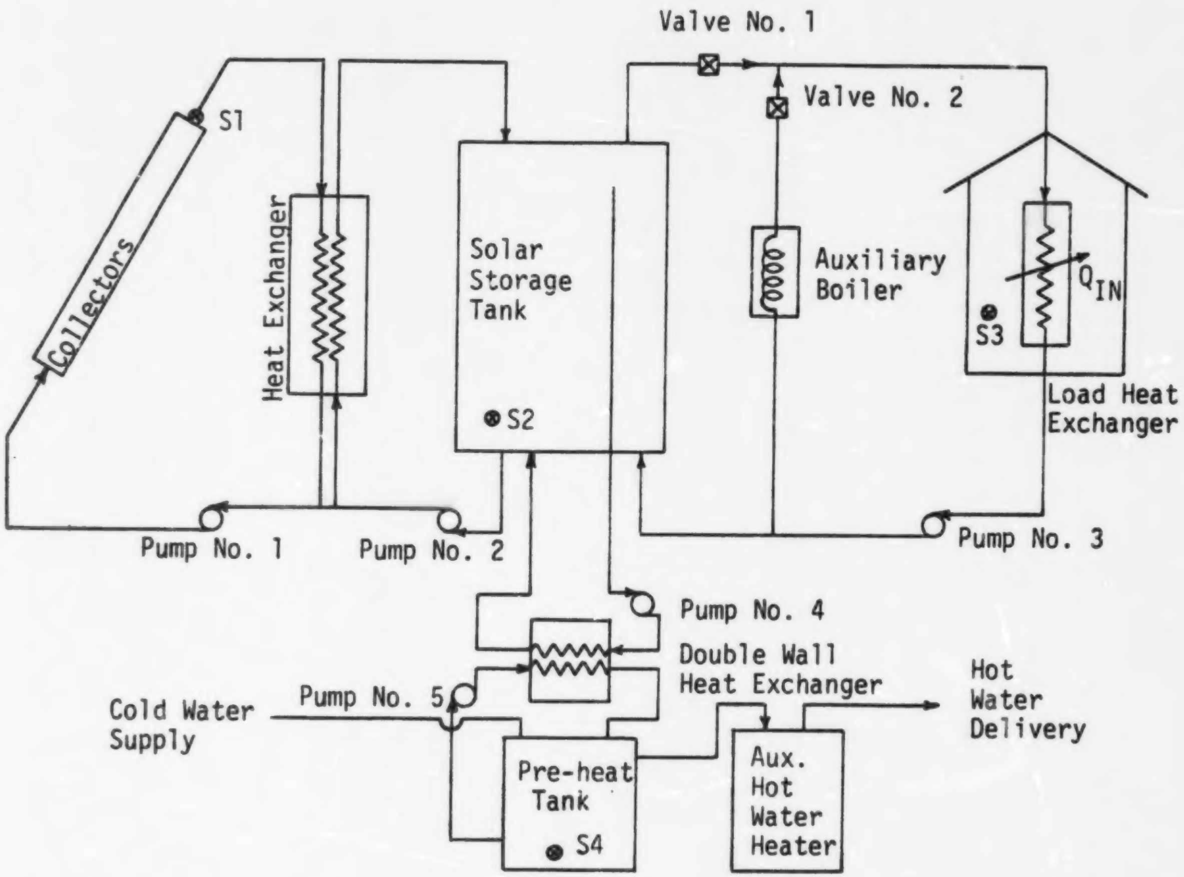

SENSORS
Functions:
Temperature Sensors
Pressure Sensors
Light Intensity Sensor
COMPARATORS
Functions:
Compare Conditions At Two Separate Sensors
Compare Condition at Sensor To A Predetermined Value
OUTPUT DEVICE
Functions:
Motor Starter
Solenoid Drive On Valve Or Damper

FIGURE 6.3.1


Valve No. 1
Valve No. 2
S1
Collectors
Heat Exchanger
Solar
Storage
Tank
S2
Auxiliary
Boiler
Q_{IN}
S3
Load Heat
Exchanger
Pump No. 1
Pump No. 2
Pump No. 3
Pump No. 4
Double Wall
Heat Exchanger
Hot
Water
Delivery
Cold Water
Supply
Pump No. 5
Pre-heat
Tank
S4
Aux.
Hot
Water
Heater

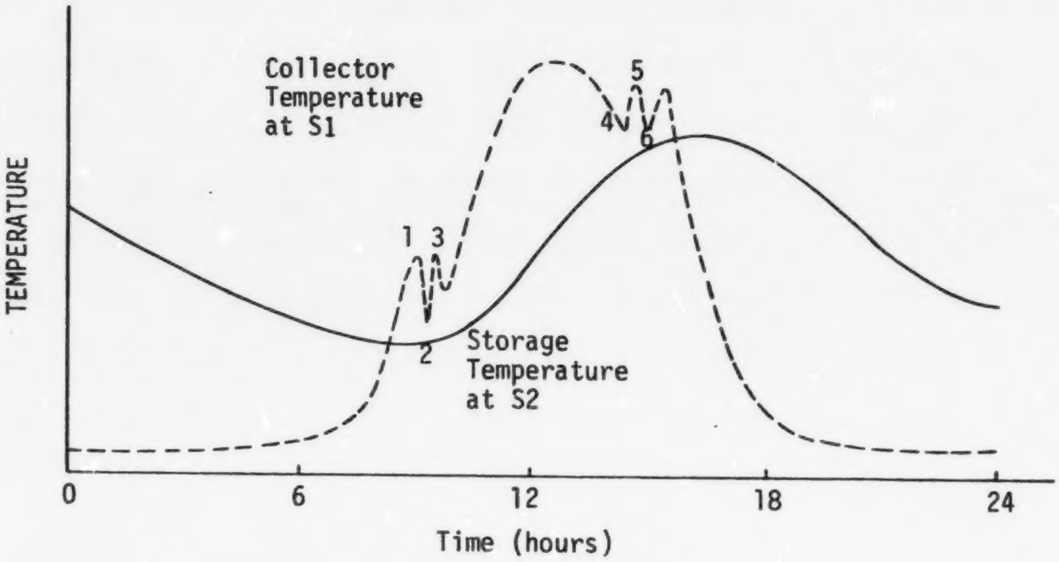
Sensor Locations in a Typical Liquid-Heating System

FIGURE 6.3.2

6.3.1 Collector To Storage Controls

When the solar collectors receive incident solar radiation, the temperature of the absorber plate rises, warming the transfer media. The collector sensor (S1 in Figure 6.3.2) senses this increasing temperature. A comparator compares the temperature of the solar collector and the thermal storage. When the temperature sensed at S1 is above that in storage at S2 by a predetermined amount, the circulation pumps (or fans) in the collector-storage loop are turned on. When the temperature difference falls to some lesser amount, the pumps turn off. The temperature differential necessary for pumps to start is typically 10°C, and the pumps cut off at a 2°C temperature differential. This deadband should be large enough to reduce on-off cycling of pump (or fan) motors, while keeping the starting temperature differential low enough so as not to significantly degrade the system efficiency. Figure 6.3.3 is an example of collector temperature profiles on a sunny day, which also indicates pump operation.

The heat transfer fluid is taken from the bottom of storage (as this is the coldest part of storage) and returned (after being heated) to the top of storage.This enhances thermal stratification in the storage and,because the coldest heat transfer fluid is being heated, collector efficiency should be maximized. The cut out temperature differential, typically 2° C, should reflect this thermal stratification and the temperature drop in the piping and ducts from the top of the collector to the storage inlet.



Typical Temperature Profiles of Collector and Storage Liquids in a Liquid-Heating System

FIGURE 6.3.3

6.3.2 Preheat Controls

Preheated domestic hot water may be taken directly from storage, or a separate preheat loop may be utilized (Figure 6.3.2).This separate preheat loop operates on the same logic as the collector-storage loop: taking hot fluid from the top of the solar storage tank, passing it through a heat exchanger and returning it to the bottom of the storage tank while cool water is taken from the bottom of the preheat tank, warmed in the heat exchanger, and returned to the top of the preheat tank. The temperature in the preheat water tank is compared to that of the solar storage to determine when the circulation pumps should operate. Preheated water is supplied to the regular water heater where it is reheated for use as necessary.

6.3.3 Space Heating Controls

Solar space heating controls must be integrated with the conventional space heating control system. One method of achieving this is by using a two stage thermostat. When the room temperature falls enough for the first stage to be engaged, heat from the solar storage is circulated through the heating system in an effort to meet the heat loads. If it cannot meet space heating loads, temperatures continue to fall until the second stage in the thermostat makes contact. When this happens, the controls function to bypass the solar space heating system and start the conventional heating system. (In the system shown in Figure 6.3.2, valve 1 closes and valve 2 opens, so that circulation is through the auxiliary space heater which has been activated to meet the space loads). When the system is operated to minimize cycling of control functions, significant temperature swings (5° C) occur in the space. Narrowing the operating bands to reduce the temperature swings promotes control cycling. One method to reduce room temperature swings is by limiting the use of storage to those times when it is above say 40° C or 50° C. When it is below this temperature, the auxiliary heater would automatically start when the first stage makes contact.

Another method is to separate the conventional and solar heating systems completely, and have the solar heating controls set slightly higher than the conventional system controls. If a demand for heat cannot be met by the solar heating system, the space temperature will fall enough to start the conventional system, which will raise space temperatures to something less than the set point of the solar heating thermostat, before turning the conventional system off. The solar heating system will continue to operate until all available stored energy is used, or until this system has met the space heat load.

This approach is one frequently used with air collectors. Air supplied at say 25° C is useful in meeting space heating loads while water storage must be 35° C or warmer to provide useful heat using a fan coil. This operating temperature difference may result in better system efficiency using air collectors over liquid collectors with a similar efficiency curve. It is possible to reduce the useful storage temperature of liquid systems through the use of heat pumps, which are now widely available.

6.3.4 Controls Selection And Placement Considerations

The location of sensors in solar energy systems is important for efficient system operation. Collector temperature should be sensed in a collector outlet at the top of the collector array. If possible, the sensor should be immersed in the solar fluid. If it cannot be directly immersed, special care should be taken to ensure that the sensors will give an accurate indication of the temperature of the solar fluid in the solar panel or of the absorber plate. The sensor on the solar collector should be able to withstand stagnation temperatures without breaking down or becoming detached from its mount.

A controller which interrupts the power supply to the pumps when low solar fluid temperatures are sensed should be located in the solar fluid line from the collectors to the heat exchanger. This would prevent freezing of the heat exchanger in the event of a controls failure (which operated the pumps despite low solar fluid temperatures).

In storage tanks, sensors should be near the bottom to detect the coldest tank temperature because the bottom water is that which is heated, and thermal stratification can be very significant.

Temperature limiters should be located at the top of the storage so that the maximum temperatures are sensed.

Pressure limiters should be set to reflect their location in the loop and they should be attached to the most pressure sensitive system component.

A method of testing control functions should be incorporated into the controls design. This will permit the operator to more readily trouble shoot the system in the event of a system failure.

A variety of good quality solid state controls designed specifically for solar applications are now marketed. These are generally quite inexpensive compared to controls systems made by mixing and matching standard HVAC components in an effort to achieve the appropriate controls logic. In addition, the solar industry designed controllers are less complex to install and are usually equipped with additional useful features.

A number of publications concerning controls for solar energy systems are listed in the references at the end of this chapter.

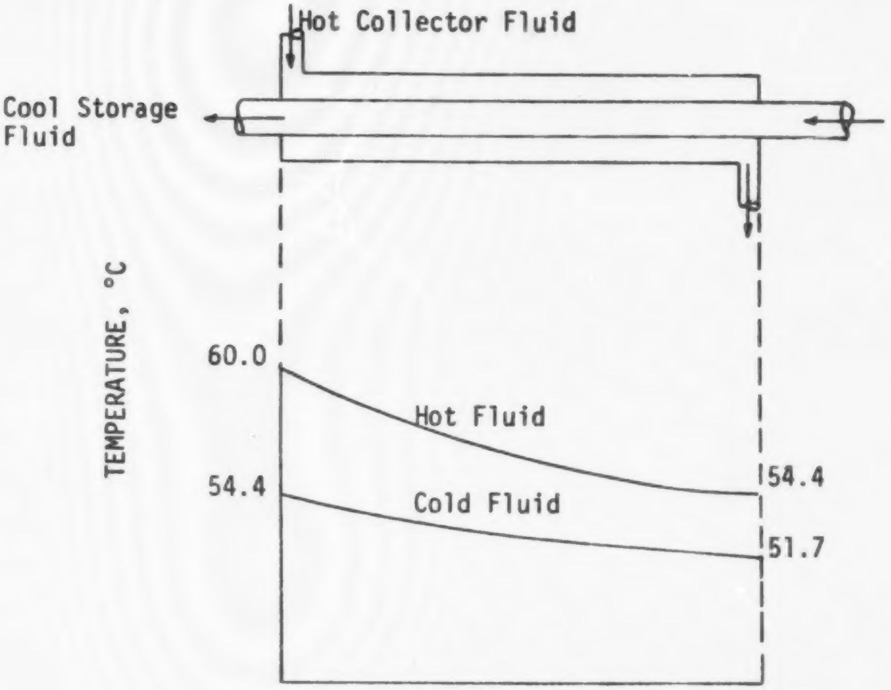
6.4 HEAT EXCHANGERS AND SOLAR FLUID

6.4.1 Introduction

Heat exchangers are available in a variety of configurations and construction materials, and can be used for a variety of applications. The selection of a heat exchanger configuration is dependent on the specific design criteria pertinent to the application at hand.

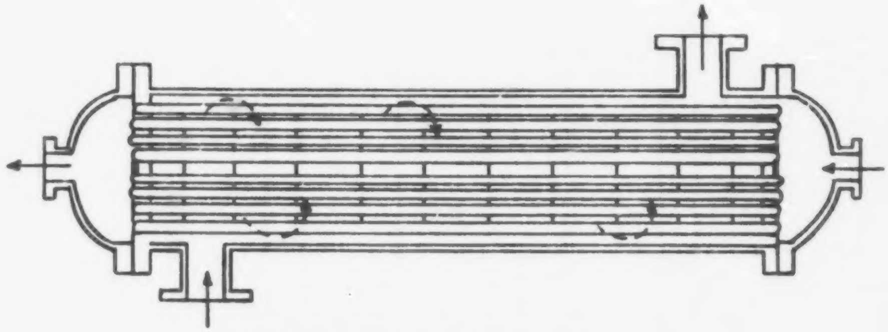
Counterflow heat exchangers, those in which the cold fluid flows in the opposite direction to the hot fluid (Figure 6.4.1), minimize the temperature differences between the outgoing hot (solar) fluid and the incoming cold (storage) fluid. Counterflow tube and shell multiple pass heat exchangers (Figure 6.4.2) are a compact and effective design suitable for use in solar energy systems. If the heat exchanger is used to heat domestic service water, double-walled heat exchangers will eliminate the possibility of solar fluid contaminating the domestic water and may be required by the plumbing code authority. A number of double-walled heat exchanger configurations are shown in Figure 6.4.3. There is not yet a national policy on this, and requirements vary from jurisdiction to jurisdiction. One standard for selecting the type of heat exchanger involves rating solar fluids according to their hazard (toxicity), and the possibility of a solar fluid leak going undetected. The combination of these hazards determines the type of heat exchanger that must be utilized for domestic water heating applications. Tables 6.4.1 through 6.4.4 highlight this approach using the guidelines set out in NRC's "Potable Water Protection in Domestic Hot Water Heating System" December, 1979, Contract #079-010.

The following sections outline considerations which can affect the safety of solar DHW heating systems.


Hot Collector Fluid
Cool Storage Fluid
TEMPERATURE, °C
60.0
54.4
Hot Fluid
Cold Fluid
54.4
51.7

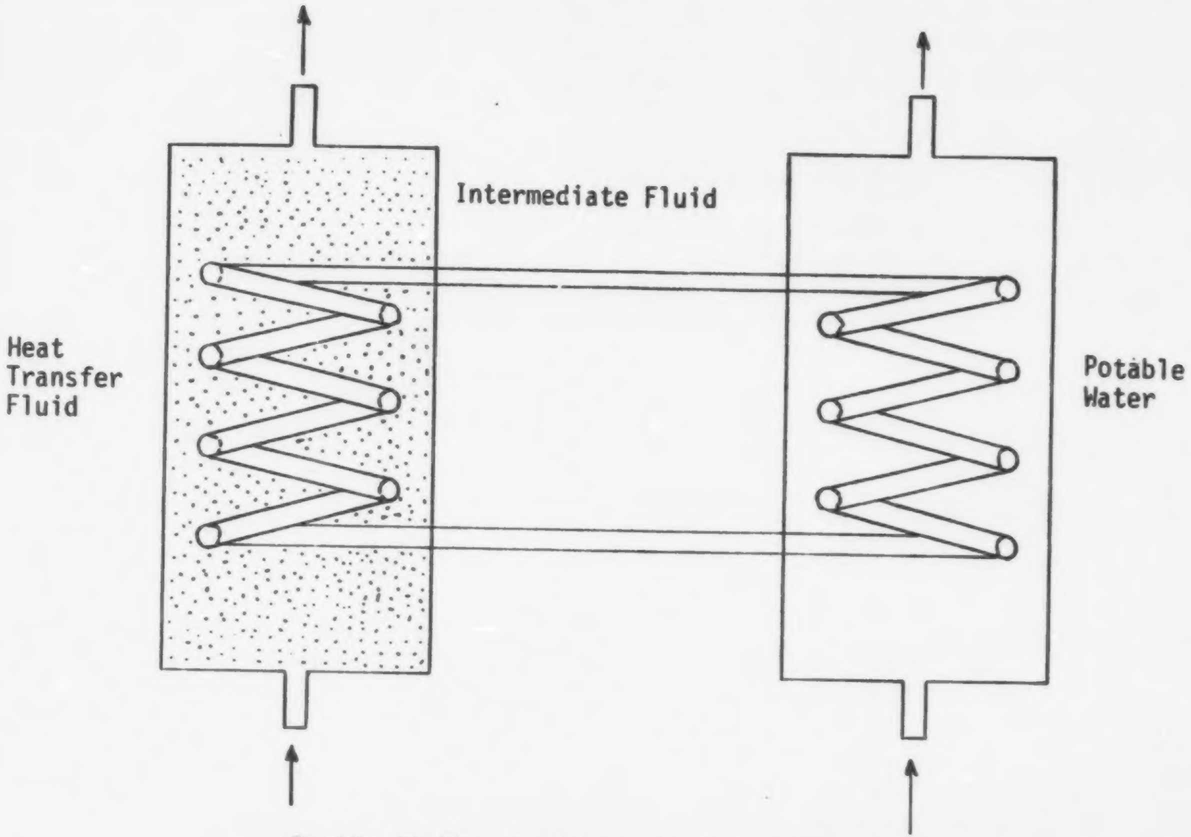
Single-Pass Counterflow Shell-and-Tube Heat Exchanger

FIGURE 6.4.1

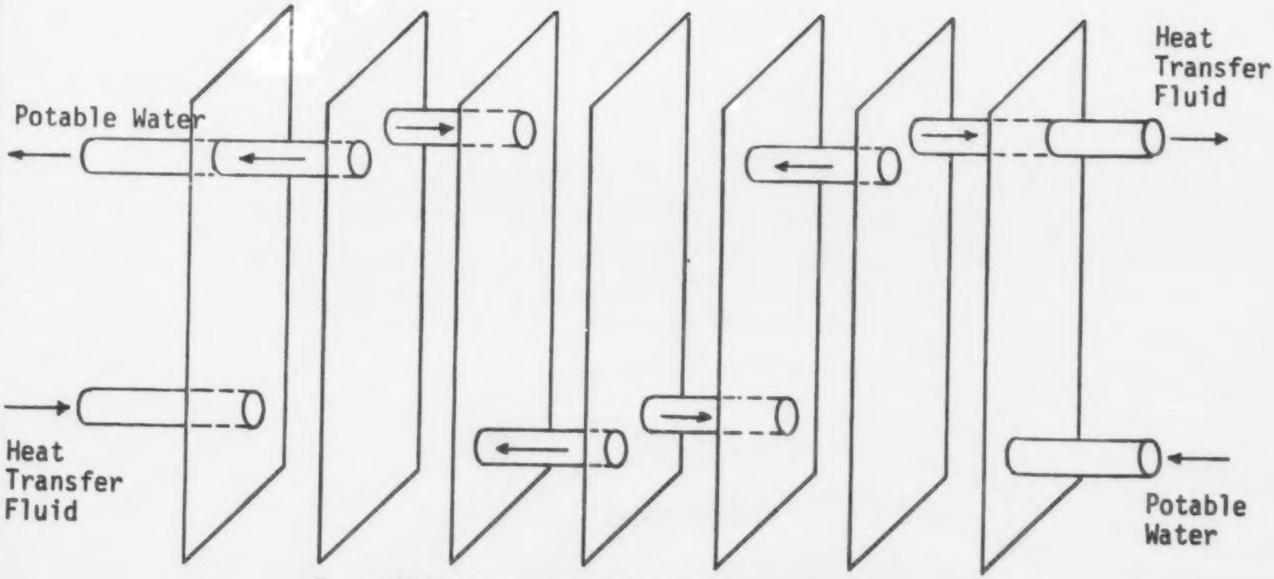


Multiple Tube Heat Exchanger

FIGURE 6.4.2

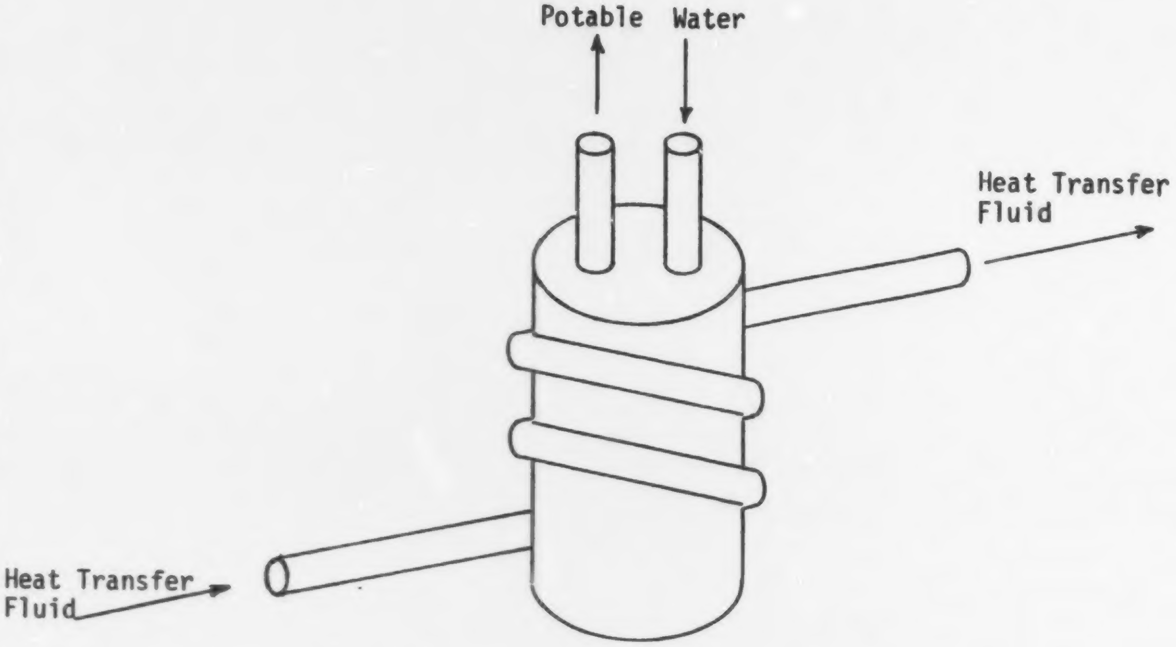

Intermediate Fluid
Heat Transfer Fluid
Potable Water

Double Shell and Tube Heat Exchanger

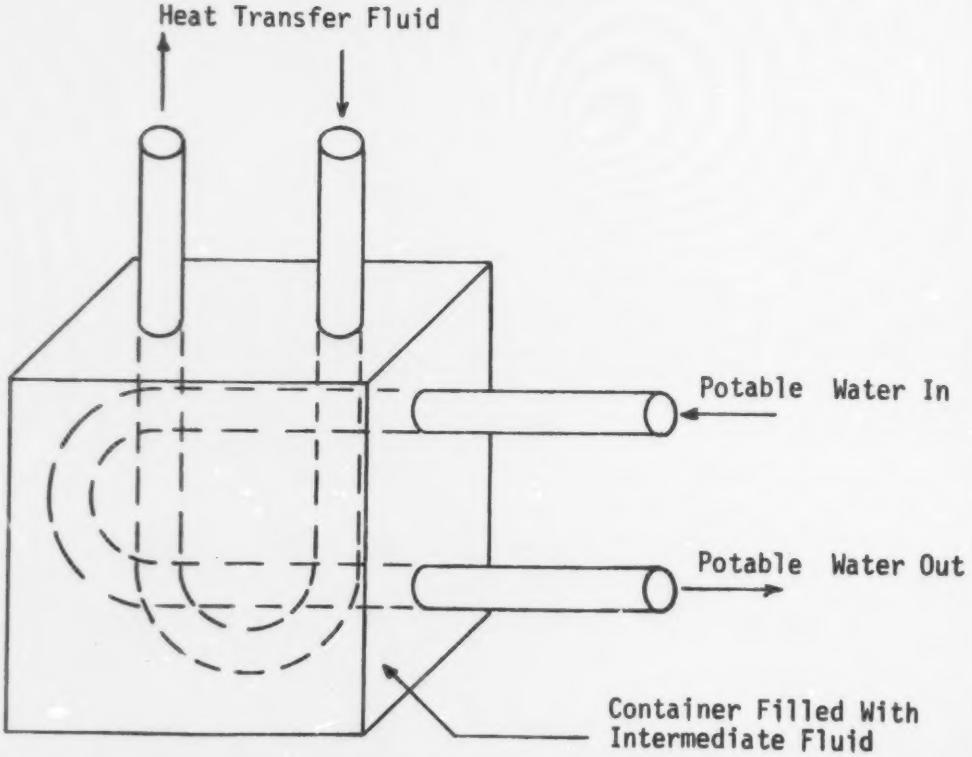

Potable Water
Heat Transfer Fluid
Heat Transfer Fluid
Potable Water

Expanded View Plate Heat Exchanger

FIGURE 6.4.3


Potable Water
Heat Transfer Fluid
Heat Transfer Fluid

Tube-On-Shell Heat Exchanger (Intimate Contact)


Heat Transfer Fluid
Potable Water In
Potable Water Out
Container Filled With Intermediate Fluid

Shell and Double Tube Heat Exchanger

FIGURE 6.4.4

TABLE 6.4.1

HEAT TRANSFER LIQUIDS AND HEAT SINK LIQUIDS CLASSIFICATIONS

CLASS A.a.	WATER SUITABLE FOR DRINKING.
CLASS A.b.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 1 AND CONTAINING A SUITABLE INDICATOR.
CLASS B.a.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 1.
CLASS B.b.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 2 CONTAINING A SUITABLE INDICATOR.
CLASS C.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 2.
CLASS D.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 3.

LIQUIDS WITH A GOSSELIN TOXICITY RATING OF GREATER THAN 3 SHALL NOT BE USED WITH ANY POTABLE WATER SYSTEM.

TABLE 6.4.2

LIQUID-TO-LIQUID HEAT EXCHANGER CLASSIFICATIONS

CLASS 1.	SINGLE WALL HEAT EXCHANGERS.
CLASS 2.a.	SINGLE WALL HEAT EXCHANGERS WITH AN ADDITIONAL LEVEL OF PROTECTION.
CLASS 2.b.	DOUBLE WALL HEAT EXCHANGERS.
CLASS 3.	DOUBLE WALL HEAT EXCHANGERS WITH AN ADDITIONAL LEVEL OF PROTECTION.
CLASS 4.	DOUBLE WALL HEAT EXCHANGERS WITH A VISUAL MEANS OF LEAK DETECTION.

TABLE 6.4.3

GOSSELIN TOXICITY RATINGS OF SOME COMMONLY USED SOLAR FLUIDS

RATING	MEANING	EXAMPLES
0	NONTOXIC	POTABLE WATER
1	PRACTICALLY NONTOXIC	PROPYLENE GLYCOL
2	SLIGHTLY TOXIC	ETHYLENE GLYCOL ETHYL ALCOHOL (BORDERLINE 2TO3)
3	MODERATELY TOXIC	PHENYL ETHER-DYPHENYL MIXTURE (DOWNTHERM A*)
4	VERY TOXIC	
5	EXTREMELY TOXIC	

---

* COMMON TRADE NAME.

TABLE 6.4.4

PERMISSIBLE HEAT EXCHANGER - HEAT TRANSFER FLUID COMBINATIONS

HEAT TRANSFER FLUID CLASS	HEAT EXCHANGER CLASS			
	1	2	3	4
A	P	P	P	P
B	N	P	P	P
C	N	N	P	P
D	N	N	N	P

N - NOT PERMITTED

P - PERMITTED

6.4.2 Reducing The Possibility Of Leakage

Careful component design and selection will reduce the potential for component failure which will result in leakage of solar fluid. Component failure may be a result of mechanical failures such as faulty welding or brazing at heat exchanger seams and joints, gasket failures or component failure due to excessive thermal shock. Corrosion in systems using reactive metals can be severe. The pH of the solar fluid, temperature, dissolved oxygen levels, the concentration of dissolved chemicals, and junctions between dissimilar metals can all rapidly accelerate corrosion. High flow velocities through system components will result in erosion, especially where flow directions change, as they do in heat exchanger U tubes. If careful attention is paid to the selection of the materials used in the system, and the design and construction of the components, the risk of leakage can be minimized.

6.4.3 Reducing Potential For Leakage Between Solar Fluid And DHW Supply

The possibility of leakage occurring between the solar fluid and the water being heated can be most effectively eliminated by reducing the potential points of cross-flow. Typically, the only place where cross-contamination can occur in a dual fluid solar energy system is in the heat exchanger.

Double-walled heat exchangers, if correctly designed and installed, can virtually eliminate the possibility of cross-contamination. There is, however, a cost/performance penalty of using double-walled heat exchangers instead of single-walled units. Experience indicates that performance information on double-walled heat exchangers is not readily available, and selection is limited.

An alternative to eliminating the potential for cross-contamination due to leakage is to attempt to restrict the direction of any leakage. By designing the solar energy system to operate with lower pressures in the solar fluid loop than in the domestic water loop, any leakage that does occur should occur from the domestic water into the solar fluid. The installation of a properly sized expansion tank and properly set pressure limiting devices will ensure that the pressure of the solar fluid is lower than operating pressures in the water system.

A failure to maintain adequate pressure in the domestic water loop could lead to contamination, unless some measure to prevent it is provided. One safeguard is a differential pressure trip which automatically drains the solar fluid loop if its pressure exceeds that on the water side. A second is to let any leak cause a blowdown, as described in the following section.

6.4.4 Detection

If a leak occurs, the ease of its detection is important. Leakage across the heat exchanger would occur from the higher pressure fluid to the lower pressure fluid. Normally the domestic water loop pressure would exceed the solar fluid pressure. Therefore, leakage would be into the solar fluid, raising the pressure in the solar fluid loop and diluting the solar fluid. This would show up as higher than normal pressure in the solar loop, which should result in solar fluid blowing down. Reduced solar fluid concentrations may make the collectors susceptible to freezing damage.

For leakage to occur into the domestic water, the pressure in the solar fluid loop must exceed the pressure in the water loop. This could occur if water pressures are drastically reduced or if component failures cause excessive pressure buildups in the solar fluid loop.

The inclusion of a dye or a mercaptan in the solar fluid would stain or odour the water if solar fluid leaks into the water supply. Some commercially available solar fluids contain a concentrated food dye. In any case, users of the water should be informed as to the warning signs.

6.4.5 Solar Fluid Toxicity

The solar system designer has some latitude in specifying solar fluids for a project. Although nontoxic fluids are available, heat transfer fluids should always be regarded as nonpotable. A nontoxic solar fluid may become toxic due to chemical additives and inhibitors, from impurities picked up or dissolved from the solar loop or as a result of decomposition due to impurities, aging, or high operating temperatures. Further, there is always a possibility that a nontoxic fluid will be replaced with a toxic one in the future. The fill point of the solar loop should be labelled, clearly specifying the solar fluid to be used in the system and the expected life of the solar fluid.

The use of the solar heated domestic water is significant in assessing the potential hazard. Normally, domestic hot water is not consumed, but is used for washing and bathing purposes. This greatly reduces the danger as the allowable concentration level for skin contact is generally orders of magnitude greater than allowable concentration levels in drinking water. Obviously if only nontoxic solar fluids are used, the toxicity hazard does not exist. Nontoxic or low toxicity solar fluids such as propylene glycol, silicone fluids, or paraffinic oils are recommended for potable water systems if the possibility of water-solar fluid cross-contamination exists.

6.5 SIZING A HEAT EXCHANGER FOR A DUAL FLUID SOLAR ENERGY SYSTEM

The purpose of the heat exchanger is to transfer the heat energy absorbed by the solar fluid in the collectors to the thermal storage. If the heat exchanger is too small, insufficient heat will be removed from the solar collector system and the leaving exchanger liquid temperature will be high. This will cause a high entering liquid temperature into the solar collector. High liquid temperature in the collector will reduce collection efficiency. Given an adequately-sized heat exchanger, the liquid temperature rise in the solar system will be moderate, leading to operation of the collection system at maximum obtainable efficiency.

The design of a glycol collector subsystem must consider a "trade-off" between the solar collector area and the glycol to water heat exchanger surface area. For the same rate of solar heat input into storage, an undersized heat exchanger will require a much larger collector surface area as compared with an adequately-sized heat exchanger. Since collector surface area is very costly compared with heat exchanger surface area, the tradeoff should be towards generous sizing of the exchanger in order to maintain collection efficiency, thus reducing collector area for equal solar heat input into storage.

The following section does not attempt to explain heat transfer in such detail as to allow the solar system designer to design the required glycol to water heat exchanger. The subject is too complex for treatment here. However, it is intended to provide an overview of heat exchanger sizing in a glycol-cooled collector system so that the solar designer can exercise judgement in terms of exchanger specification, leaving the final design and sizing to experts in this field.

Liquid to liquid heat exchangers used in solar water heating systems experience different thermal operating conditions than do heat exchangers used in boiler water heating systems. Manufacturers of heat exchangers usually use computer programs to select heat exchangers for customers.These programs make the selection based upon one set of (constant) conditions, that the heat exchanger will experience, as specified by the customer. However, in hydronic solar systems, the operating conditions of the heat exchanger are not constant. The diurnal climatic fluctuations and load variations cause significant changes in the hot fluid temperature throughout the day. If a heat transfer fluid is used that has a viscosity that changes significantly over the temperature range typically experienced in a flat plate solar collector, 15 - 80°C (59 - 180°F), as the viscosities of solutions of ethylene and propylene glycol do, special care must be taken when selecting the heat exchanger design parameter values.

Changes in viscosity affect heat transfer in two ways:

1. Increased viscosity reduces fluid flow, thereby decreasing the turbulence of the fluid, and lowering the conductance of a heat exchanger.
2. Increased viscosity causes a "thickening" of the laminar flow boundary layer at the wall of a heat exchanger also lowering the conductance of a heat exchanger.

The effects of increased viscosity on heat exchanger conductance can be countered by increasing the flow rates of the fluids. However, this solution is not always practical because of the increased pumping costs.

In summary, a heat exchanger for a solar water heating system, chosen by using a heat exchanger manufacturer's computer program, will be incorrectly sized unless the values of the thermal sizing parameters are consistent with the operating conditions actually experienced by the heat exchanger.

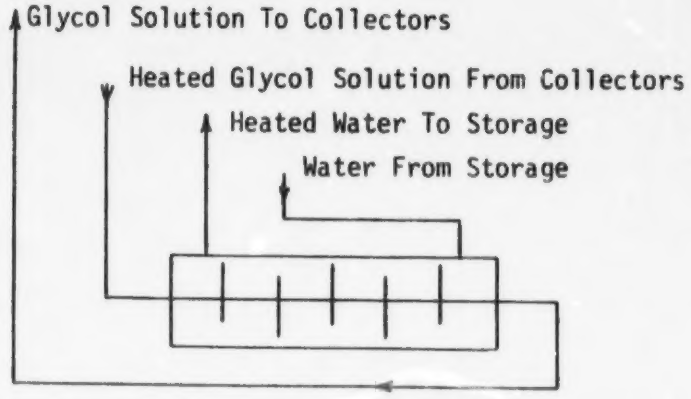
Three basic types of glycol to water heat exchanger units are described for use with glycol-cooled collector subsystems. They are illustrated in Figure 6.5.1.

The counter flow straight tube heat exchanger (A in Figure 6.5.1) will be used to describe approach temperature difference as this factor influences heat exchanger sizing and consequent collector operating efficiency.

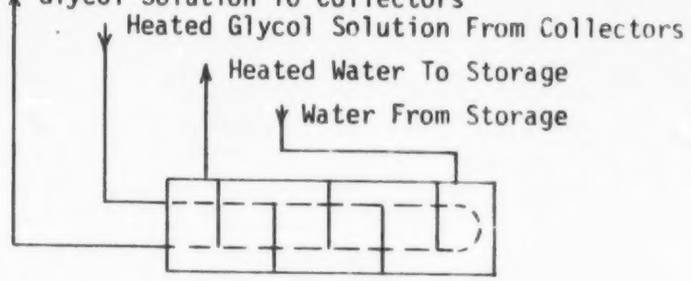
Approach temperature difference for a heat exchanger is the difference in temperature between the incoming heating fluid and the leaving heating fluid. Figure 6.5.2 illustrates the approach temperature difference (ΔT) for a glycol to water solar heat exchanger. The exchanger is designed for a 10°F (5.5°C) "approach".

The approach ΔT used for design of the heat exchanger has great significance for glycol-cooled collector systems because of its effect on the solar panel temperature required to meet a stated heat exchanger load. The higher the design approach ΔT, the higher the mean solar panel liquid temperature required to meet the load.

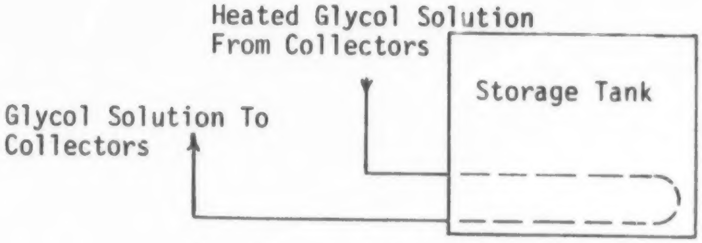
Figure 6.5.3 illustrates the effect of heat exchanger design approach ΔT on required solar collector liquid temperatures for a fixed load.


Glycol Solution To Collectors
Heated Glycol Solution From Collectors
Heated Water To Storage
Water From Storage

A: Counterflow Straight Tube Heat Exchanger


Glycol Solution To Collectors
Heated Glycol Solution From Collectors
Heated Water To Storage
Water From Storage

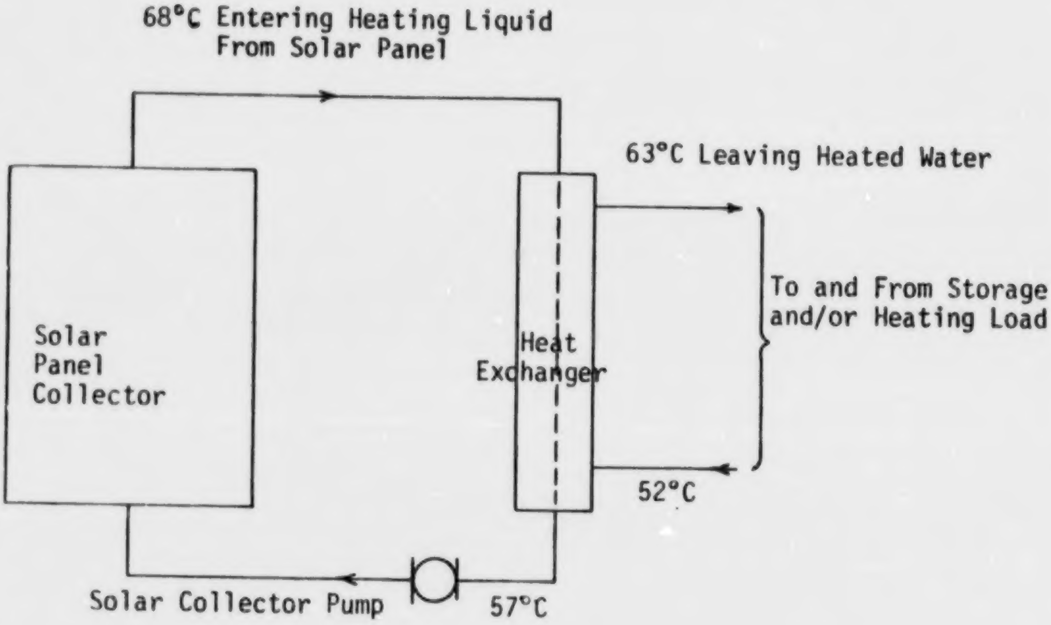
B: "U" Tube Heat Exchanger


Heated Glycol Solution
From Collectors
Storage Tank
Glycol Solution To
Collectors

C: Immersed Storage Tank Heat Exchanger

Heat Exchangers Used for Glycol Solution
Filled Solar Collector Systems

FIGURE 6.5.1


68°C Entering Heating Liquid
From Solar Panel
63°C Leaving Heated Water
To and From Storage
and/or Heating Load
Solar
Panel
Collector
Heat
Exchanger
52°C
Solar Collector Pump
57°C

Heat Exchanger Designed for 5° Approach (68°-63°-5°)

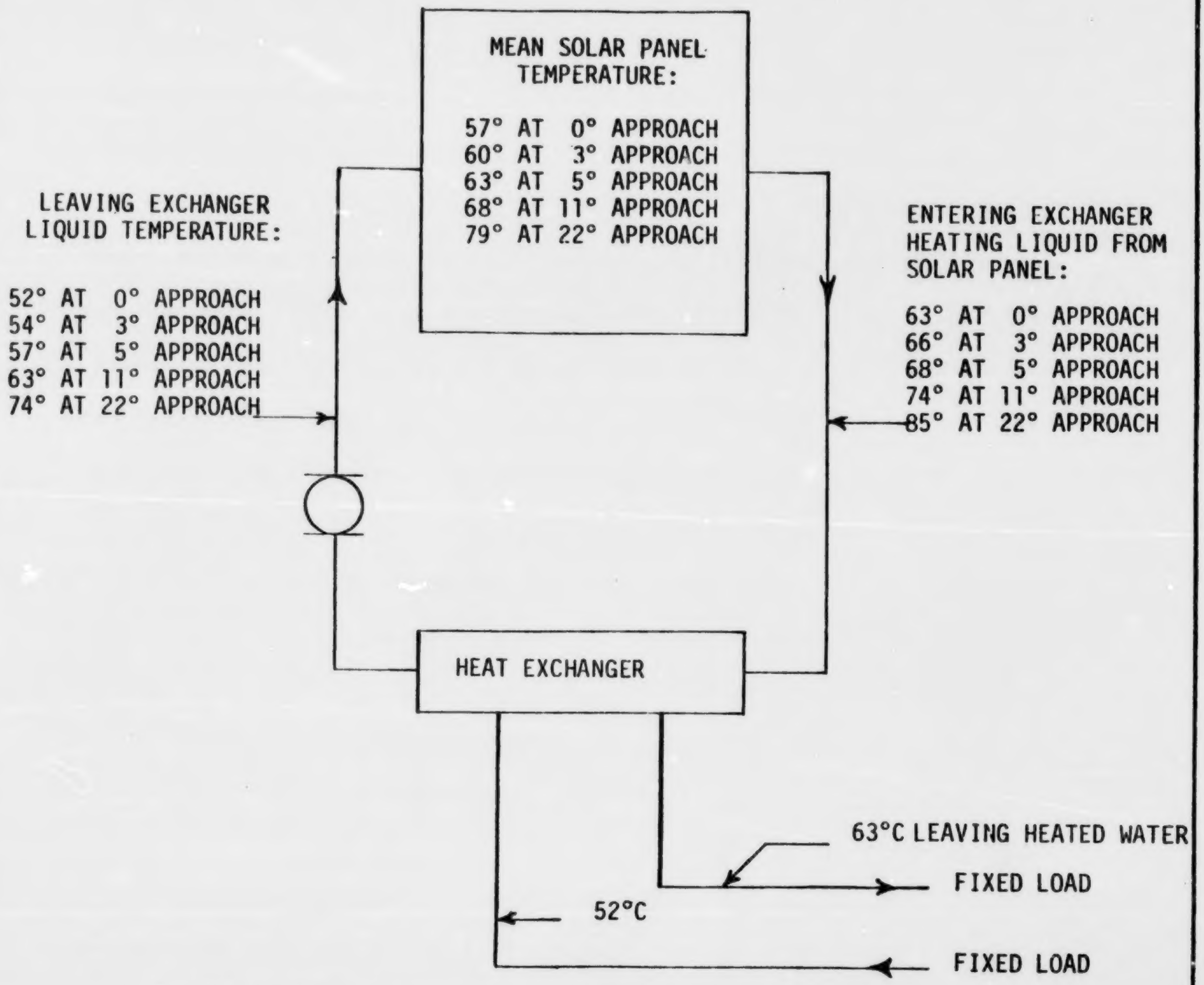
FIGURE 6.5.2

The 0° approach shown in Figure 6.5.3 and in Table 6.5.1 refers to a collector system which has no heat exchanger (water drain back system) and serves as a base for evaluation of the use of approach ΔT for the specification of solar glycol to water exchangers.

The glycol to water heat exchanger used in solar systems should be base sized for a 2.8°C to 8.3°C (5°F to 15°F) approach, with 5.5°C (10°F) as the recommended value. The reason is that exchangers sized to less than a 2.8°C approach will provide very little increased collector panel performance for a comparatively large increase in exchanger size and cost. Heat exchangers sized for greater than an 8°C approach will reduce panel performance excessively (relative to the decreased cost of the exchanger).

It should be noted that approach ΔT is a rough comparative measure of heat exchanger size. A heat exchanger selected for a 22°C (40°F) approach will be about 1/4 the size of an exchanger selected for a 4.5°C (10°F) approach. The difference in cost will generally not be proportionate. Considering the relatively high installed cost of the solar collector panels, it would be most unwise to sacrifice panel performance for the relatively small decrease in exchanger cost established by design to a large approach ΔT.

The effectiveness of a heat exchanger, ε, compares the actual heat transfer rate, to the thermodynamically limited maximum heat transfer rate, as would be realized only in a counterflow heat exchanger of infinite heat transfer area, and may be determined by the following formulae:


MEAN SOLAR PANEL TEMPERATURE:
57° AT 0° APPROACH
60° AT 3° APPROACH
63° AT 5° APPROACH
68° AT 11° APPROACH
79° AT 22° APPROACH
LEAVING EXCHANGER LIQUID TEMPERATURE:
52° AT 0° APPROACH
54° AT 3° APPROACH
57° AT 5° APPROACH
63° AT 11° APPROACH
74° AT 22° APPROACH
ENTERING EXCHANGER HEATING LIQUID FROM SOLAR PANEL:
63° AT 0° APPROACH
66° AT 3° APPROACH
68° AT 5° APPROACH
74° AT 11° APPROACH
85° AT 22° APPROACH
HEAT EXCHANGER
63°C LEAVING HEATED WATER
FIXED LOAD
52°C
FIXED LOAD

Change in Mean Liquid Temperature in Solar Collectors with Change in Heat Exchanger Design Approach ΔT

FIGURE 6.5.3

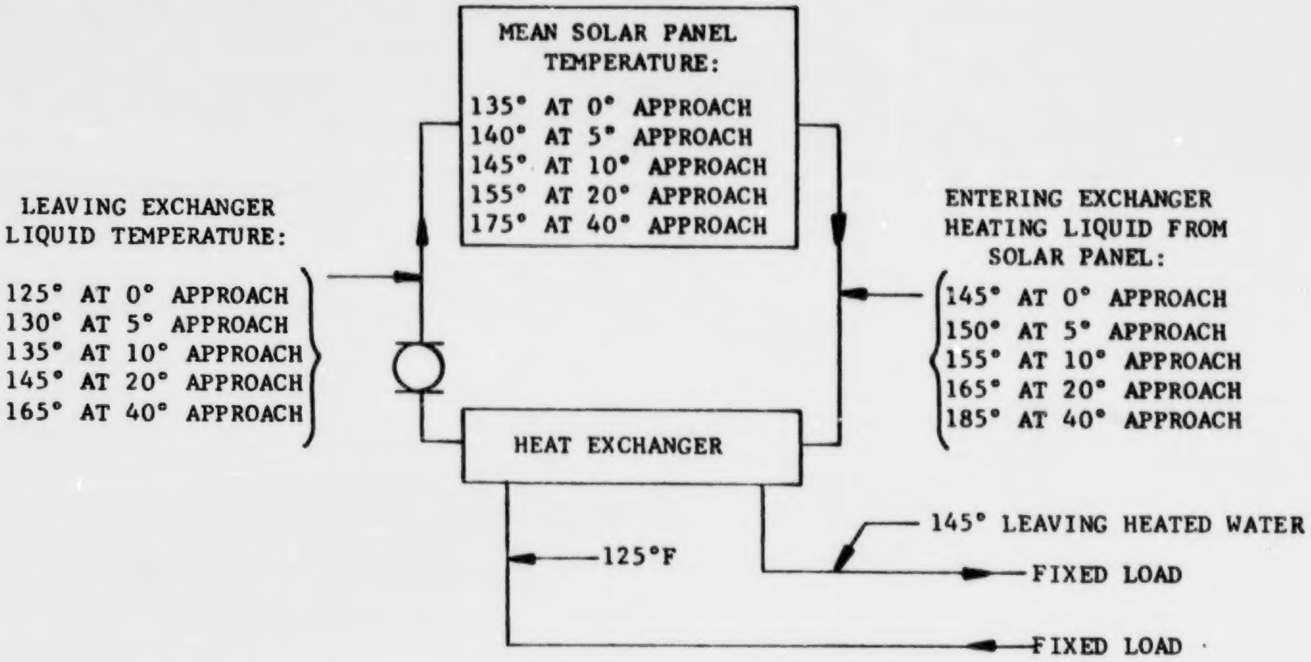

MEAN SOLAR PANEL TEMPERATURE:
135° AT 0° APPROACH
140° AT 5° APPROACH
145° AT 10° APPROACH
155° AT 20° APPROACH
175° AT 40° APPROACH
LEAVING EXCHANGER LIQUID TEMPERATURE:
125° AT 0° APPROACH
130° AT 5° APPROACH
135° AT 10° APPROACH
145° AT 20° APPROACH
165° AT 40° APPROACH
ENTERING EXCHANGER HEATING LIQUID FROM SOLAR PANEL:
145° AT 0° APPROACH
150° AT 5° APPROACH
155° AT 10° APPROACH
165° AT 20° APPROACH
185° AT 40° APPROACH
HEAT EXCHANGER
125°F
145° LEAVING HEATED WATER
FIXED LOAD
FIXED LOAD

Figure 6.5.3 Change in Mean Liquid Temperature in Solar Collectors with Change in Heat Exchanger Design Approach ΔT

The increase in mean solar panel temperature with increased heat exchanger approach temperatures will affect collection efficiency as follows for the collector described in Figure 3.2 and for 40° outdoor temperature.

TABLE 6.5.1

HEAT EXCHANGER APPROACH ΔT °F	MEAN PANEL TEMPERATURE °F	OUTDOOR AMBIENT °F	ΔT; MEAN PANEL MINUS OUTDOOR °F	COLLECTOR PANEL EFFICIENCY AT 250 B/H/FT^2 INSOL.
0° (Base)	135	40	95	33
5°	140	40	100	32.5
10°	145	40	105	32
20°	155	40	115	29
40°	175	40	135	24

$$\varepsilon = \frac{C_h\,(T_{\text{hot in}} - T_{\text{hot out}})}{C_{\min}\,(T_{\text{hot in}} - T_{\text{cold in}})} \qquad \text{(Eqn 6.5.1)}$$

$$\varepsilon = \frac{C_c\,(T_{\text{cold out}} - T_{\text{cold in}})}{C_{\min}\,(T_{\text{hot in}} - T_{\text{cold in}})} \qquad \text{(Eqn 6.5.2)}$$

Where C min is the smaller of Cc and Ch

C_c = the capacitance of the cold fluid
J/s C° (BTU/hr. F°)

C_h = the capacitance of the hot fluid
J/s C° (BTU/hr. F°)

The solar system designer will have to specify selection conditions (type of fluids, approach ΔT, expected liquid temperature and flow rates, allowable head loss, fouling factors, etc.) in order to allow the heat transfer specialist to select the lowest cost, most effective heat exchanger for the system.

When shell and tube exchangers are used, the solar system designer should allow the specialist the following options in order to obtain best performance at lowest cost.

1. The heat exchanger may be either "U" tube or counter flow straight tube. (See Figure 6.5.1)
2. The heat exchanger may be either single or multiple units or passes.
3. Glycol solution flow may be either through the tube or through the shell.
4. A maximum approach temperature of 5.5°C is recommended. While the specified approach ΔT should not be exceeded, a lower ΔT should be allowed. This will often permit the use of "off the shelf" exchangers of larger size than actually needed, but which have a lower cost than the units specifically manufactured to meet exacting specifications.
5. The maximum heat exchanger length may be specified by the specialist. Sufficient room must be allowed to permit removal of tubes from shell of heat exchanger.

6. The design temperature for the heat exchanger may be specified by the specialist. As the heat transfer characteristics of glycol solutions vary greatly with temperature, this is an important variable. The recommended design temperature is 40°C (or 104°F).

7. The specialist may specify fluid flow rates that are as high as practical to maximize heat transfer rates.

6.6 PUMPS AND PIPING

Centrifugal pumps are the best choice for pumping clean fluids against moderate pressures. They are readily available for a wide variety of capacities and pressure ranges at reasonable cost.

Pumps for solar fluids require special consideration in their selection. Mechanical seals which work for water may not be effective in preventing solar fluids from leaking. Pump specifications should make reference to the solar fluid being pumped, so that the supplier can select appropriate pump seals.

The viscosity of most solar fluids changes greatly with temperature, affecting flow rates and line pressures. The pump selected must be capable of operating without overloading over the full range of operating viscosities, and must move the required amount of solar fluid in the normal operating ranges.

Pump motors should be capable of handling slight overloads, but should not be excessively oversized. This is because electric motors under 50 kW require virtually same amount of power input, regardless of load in the normal operating range. A typical motor power-efficiency curve, illustrated in Figure 6.6.1, helps to demonstrate this concept. If the load on an electric motor is doubled from a to b, the efficiency increases from 40% to 70%, with an electrical energy consumption increase of only 14%.

To minimize heat and head losses, pipe runs should be as short as possible and all pipes, valves, tanks, heat exchangers, etc. should be

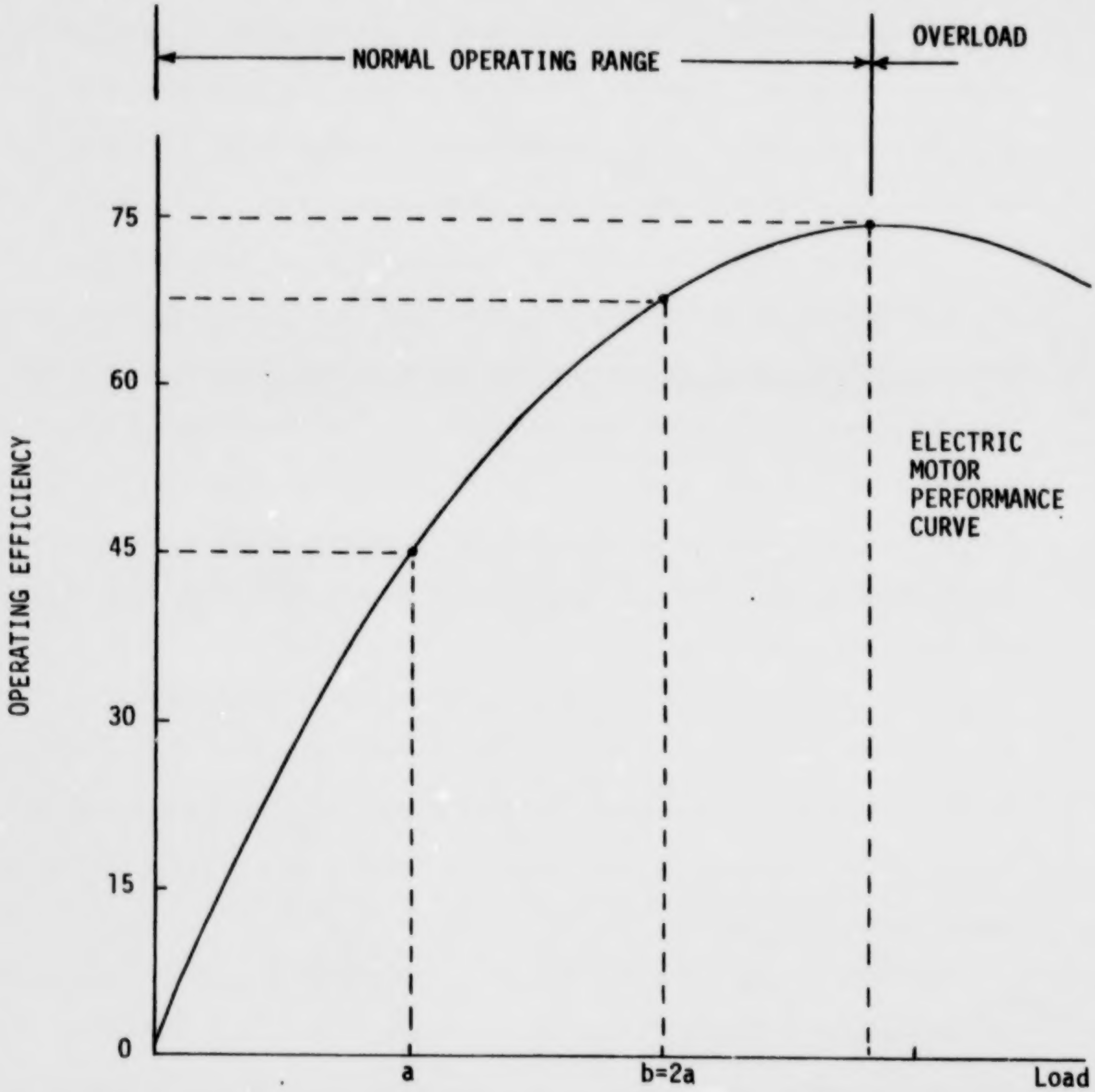

NORMAL OPERATING RANGE
OVERLOAD
OPERATING EFFICIENCY
75
60
45
30
15
0
ELECTRIC
MOTOR
PERFORMANCE
CURVE
a
b=2a
Load

FIGURE 6.6.1

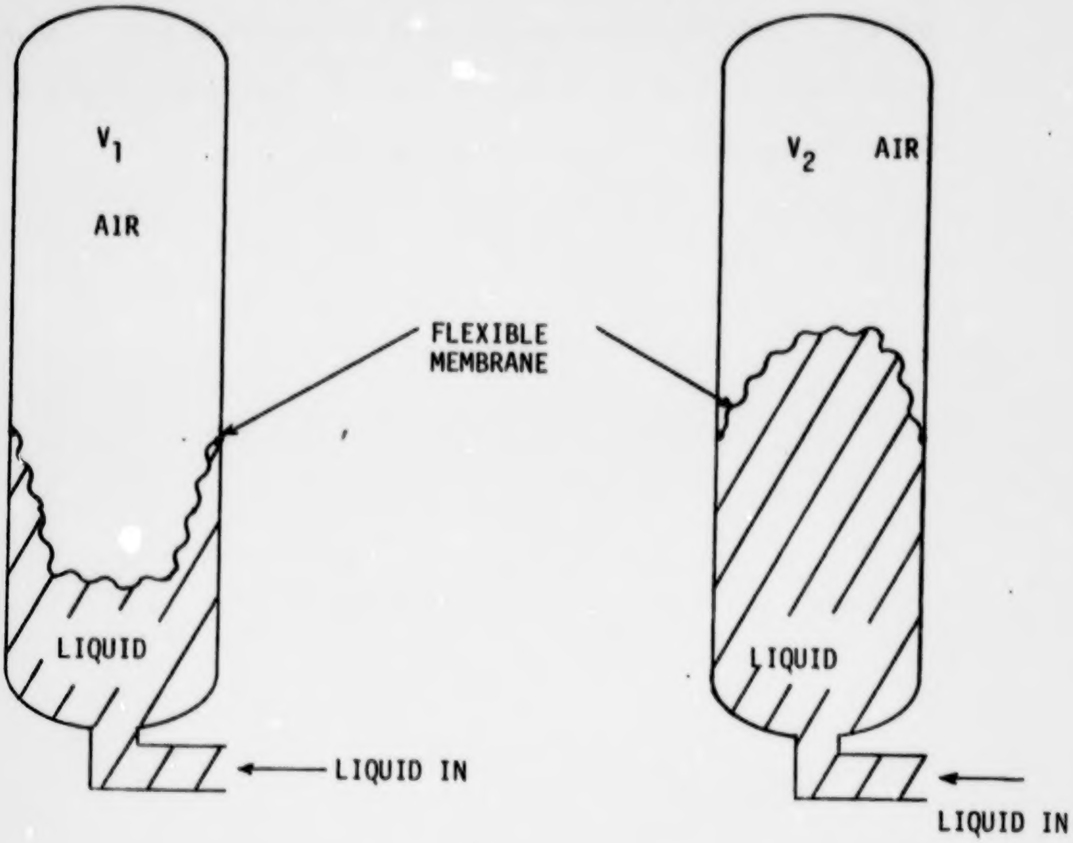
adequately insulated. The rule of thumb in sizing pipes is to select flow velocities of 1.5 meters per second or less. Higher velocities result in noise, erosion of piping and excessive head losses. Careful selection of other system components will reduce overall head losses.

Strainers should be installed upstream of pumps to reduce wear. Pressure gauges located around pumps are useful in quantifying flow through the pump when performance testing the system. They can also draw attention to unusual changes in pump performance. Similarly, temperature gauges located on the inlets and outlets of the heat exchanger are helpful in evaluating system performance and in selecting optimum temperature differences for the controls functions. There are PUSH guidelines regarding necessary monitoring systems for solar projects.

Solar fluid circulation loops experience very major temperature fluctuations, perhaps up to 200°C. The thermal expansion and contraction in piping and collector headers may be extreme, requiring the use of expansion loops, expansion compensators or flexible couplings in piping runs and between collectors.

Similarly, thermal expansion of the solar fluid must be facilitated. In unpressurized systems, freeboard or an overflow in the thermal storage tank may accommodate this expansion. In pressurized thermal storage, pressure relief valves and/or expansion tanks can absorb volume changes. Volume changes of 5% can occur in water storage fluctuating between 4°C and 100°C. Glycol solution fluctuating between 0°C and 150°C experiences volume changes of over 13%. When selecting an expansion tank for a closed loop system, the designer must know the volume change to be accommodated and the acceptable operating pressures in the fluid loop. Figure 6.6.2 shows a schematic of an expansion tank and shows the pressure volume relationships used in calculating tank size.

AIR VOLUME V = TANK VOLUME - LIQUID VOLUME


V_1
AIR
V_2 AIR
FLEXIBLE MEMBRANE
LIQUID
LIQUID
LIQUID IN
LIQUID IN

BUT

$V_2 = V_1$ - CHANGE IN LIQUID VOLUME

$$P_1V_1 = P_2\ (V_1 - \Delta V)$$
$$P_2\ \Delta V = P_2V_1 - P_1V_1$$
$$P_2\ \Delta V = (P_2 - P_1)\ V_1$$
$$V_1 = \frac{P_2\ \Delta V}{P_2 - P_1}$$

FIGURE 6.6.2

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CHAPTER 7

DURABILITY OF SOLAR COLLECTORS AND SYSTEMS

7.1 DURABILITY OF SOLAR COLLECTORS

Like all system components, solar collectors are subject to a variety of potential problems which can cause degradation of performance, reduced life and increased maintenance burdens. Some of the more common problem areas are discussed below. When evaluating collectors, the system designer should be very conscious of how manufacturers have handled these potential problems in their designs.

1. Glazing systems - A summary of properties of some common cover plate materials is given in Table 3.2.1. Glazing systems are subject to breakage from foreign object impact and thermal stresses, discolouration, embrittlement, loss of physical stability and abrasion. Breakage from foreign objects (vandalism) has not proved to be a major problem, even in public schools. Most breakage of glazing appears to be the result of excessive thermal stresses. For this reason, tempered or heat-strengthened glass is generally used where glass is specified. Plastics are less subject to breakage although Tedlar covers have been known to fail in extreme cold because they were overtightened on their frames. Discolouration of plastic glazings is caused by ultraviolet light reacting with polymer molecules. Polycarbonates such as Lexan, are particularly susceptible to this type of

damage. FRP (fiberglass) glazings generally have protective coatings or inhibitors added to provide UV protection. Fluorocarbons (PVF) such as Tedlar, are subject to embrittlement if not properly protected. Many plastics are also vulnerable to loss of physical stability under high temperature conditions. Some films, for example, can sag enough to actually touch the absorber plate. In very windy areas, abrasion due to wind-borne dust and dirt can also be a problem.

2. Absorber Plates - Certainly the greatest problem with absorber plates in liquid collectors is the possibility of corrosion. Corrosion in collectors and systems is discussed in more detail in Section 6.3. Since most absorber plates are metal they are also subject to problems with thermal expansion which, under extreme conditions, can cause buckling of up to 3 cm. Thermal expansion can also break soldered joints. Blockage of flow channels due to foreign objects is a remote possibility but can be handled with proper filtration.
3. Absorber coatings - Absorber coatings may be subject to degradation and discolouration due to extreme heat or to moisture. Selective surfaces are vulnerable to greying and bleaching. There is also the potential of deficiencies in the coating application. For example, thick coatings can crack or outgas under high temperature conditions. A summary of absorber coating properties is given in Table 3.2.2.

4. Insulation - Problems with insulation include outgassing from the binder or from foam insulation, moisture degradation and dimensional instability. Loose fill insulation will settle and is seldom used. Polyurethane (foamed in place) is unstable above 50° C and has caused absorber plate buckling.
5. Condensation - Condensation generally occurs when water vapour inside the collector box condenses on a cool cover plate during periods of low ambient temperatures. It can seriously degrade collector performance, ruin the absorber surface finish and cause or accelerate corrosion. Therefore all collectors should contain some type of condensate control mechanism. Three types of solutions are generally used: desiccants (usually located in breather hole), hermetic sealing of the collector (with the air inside being purged to remove any moisture) and weep holes and vents (which must be kept free and unclogged at all times).
6. Air Leakage - Proper sealing is of extreme importance to collector performance. Seals can dry out, embrittle or otherwise degrade in ultra-violet exposure. Butyl seals are very common, but experience indicates poor results; silicone seals give the best results but are very expensive. Leakage is primarily a problem with air collectors. Interestingly enough, it can actually improve system performance under certain conditions. For example, if the leakage is into the collector and is less than the fresh air requirements of the building (and is therefore required anyway), then the average operating temperature of the collector

will be lowered and hence the efficiency increased. For preheating systems, leakage is not a problem.

7.2 DURABILITY OF SOLAR ENERGY SYSTEMS

Many of the historical problems which have plagued solar energy systems have been the result of poor design and construction procedures which often ignore commonly accepted practices. With the industry now beginning to mature , many of these problems are being eliminated.

1. Transport systems - The most common problems are inadequate insulation in liquid systems and leakage in air systems. Since both are difficult to correct after the sytem has been installed, they should be carefully attended to prior to final systems commissioning. Insulation levels should be higher than normal to reflect the higher cost of heat from solar energy systems. Air systems should be carefully checked to seal all leaks before the insulation is applied. Leakage past outdoor air dampers has been known to freeze heat exchangers. Flexible plastic and rubber connectors often fail due to temperature, ozone, UV degradation or poor clamping.
2. Storage - The storage systems are generally trouble-free. There may be some biological growth in both air and liquid systems. Inadequate insulation and leakage in very large storage tanks has also been reported.

3. Controls - Controls are often too expensive, complicated and, as a result, trouble-prone. Attempts at oversophistication result in too many modes of operation with extensive numbers of devices to control. Improperly adjusted setpoints will cause cycling and result in unnecessary wear on the system. Temperature sensors, especially those in the collectors are sometimes improperly located with no provision for replacement. (Many solar collectors do not have provision for sensors.) Valves and dampers that close too slowly will waste energy, while solenoid valves that close too quickly cause water hammer. Thermosiphoning may result if reverse flow valves or dampers are not included. Monitoring equipment to diagnose faults or monitor performance is often lacking.

4. Installation - Ice dams on roofs can be caused by collector shading. Increased snow and wind loads on roofs can be created by a collector array and must be accounted for at the design stage. Note that a collector tilt angle of approximately 65° is required to shed snow. Problems may also arise from inadequate room for snow run-off or inadequate sealing of penetrations in the building membrane. The design should also leave room for movement of leaders and modules. Cranes are often required for collector installation. Mounting structures can be very expensive, and the use of a vertical wall for the collectors should be carefully considered.

5. Maintenance - Provision for maintenance is a necessity. A good set of drawings and operation instructions must be provided with the system. Remember that the building's maintenance man or operating engineer may be totally unfamiliar with solar energy systems.

A collection of design and installation guidelines is provided below.

1. The Solar Structure

DO provide unobstructed southern exposure.
DO insulate the structure very well and seal off air leaks, remember that conservation is the cheapest "source" of energy.

2. The Collector Mounting

DO mount the collector lower edge well above the average snow depth (roof mounting is ideal), to allow snow run-off.
DO retain a waterproof roof under the collector.
DO use 2 to 5 cm stand offs under the collector to prevent rot under collector.
DO use temporary scaffolding or fork lift to install the collectors.
DO mount the collector with the proper direction running vertical.

3. Collector Piping

DON'T use plastic or PVC piping without careful consideration. Many types only last 5 years. Swimming pool heating systems are an exception.
DON'T use rubber hoses with screw-type hose clamps.
DO use an expansion tank on the closed loop (collector loop).
DO use a sight glass on the expansion tank.
DO make the collector drainable and fillable.
DO use expansion bellows or "U"-tubes for thermal expansion.
DO use highest grade hydraulic hose or equivalent if flexible hoses are necessary.
DO install automatic or manual air bleed valves at the top of the tank.
DON'T plumb in inverted "U" traps which will trap air in the lines.
DO provide means for monitoring the fluid pH and freezing point.

4. System Start-up and Check-out

Do leak-check with compressed air to 207 kPa and soap spray on each solder joint.
DO mount (at least temporarily) a pressure gauge 0 - 207 kPa or higher.
DO fill the system from the bottom up with an auxiliary pump.

5. Storage System

DON'T oversize storage.
DO provide separate footing for storage if necessary.
DO use foam storage insulation if the storage tank is to be buried, i.e. make the insulation waterproof. FRP tanks are O.K. but generally more expensive.

6. The Back-Up Heating Plant

DON'T put the solar heat coil downstream of the furnace.
DO put the fan coil in front of the furnace blower.
DON'T overheat the blower motor.

7.3 CORROSION

The five most common types of corrosion and their methods of prevention and control are discussed below.

1. Galvanic Corrosion

This type of corrosion can result when two dissimilar metals come into contact in the presence of an electrolyte, that is, a suitable liquid. The corrosion on the less resistant (or less noble) metal is increased while the rate of attack on the more resistant metal is decreased. The less noble metal is known as the anode and the more noble metal the cathode. Metals can be listed in terms of their resistance to corrosion, as in the electromotive force or galvanic series. A galvanic series of some commercial metals and alloys is shown in Figure 7.3.1.

Galvanic Series of Some Commerical Metals and Alloys in Seawater.

(Anodic or active end)
Magnesium and magnesium alloys
Zinc
Commercially pure aluminum (1100)
Cadmium
2024 aluminum (4.5Cu, 1.5Mg, 0.6Mn)
Steel or iron
Cast iron
Chromium steel (active) 13% Cr
Ni-Resist (high-Ni cast iron)
18-8 stainless steel (active)
18-8-Mo stainless steel (active)
Lead-tin solders
Lead
Tin
Nickel (active)
Inconel (active: 80Ni, 13Cr, 7Fe)
Hastelloy B (60Ni, 30Mo, 6Fe, 1Mn)
Chlorimet 2 (66Ni, 32Mo, 1Fe)
Brasses (Cu-Zn)
Copper
Bronzes (Cu-Sn)
Cupronickels (60-90Cu, 4-10Ni)
Monel (70Ni, 30Cu)
Silver solder
Nickel (passive)
Inconel (passive)
Chromium steel (passive)
18-8 stainless steel (passive)
18-8 Mo stainless steel (passive)
Hastelloy C (62Ni, 17Cr, 15Mo)
Chlorimet 3 (62Ni, 18Cr, 18Mo)
Silver
Titanium
Graphite
Gold
Platinum
(Cathodic or noble end)

FIGURE 7.3.1

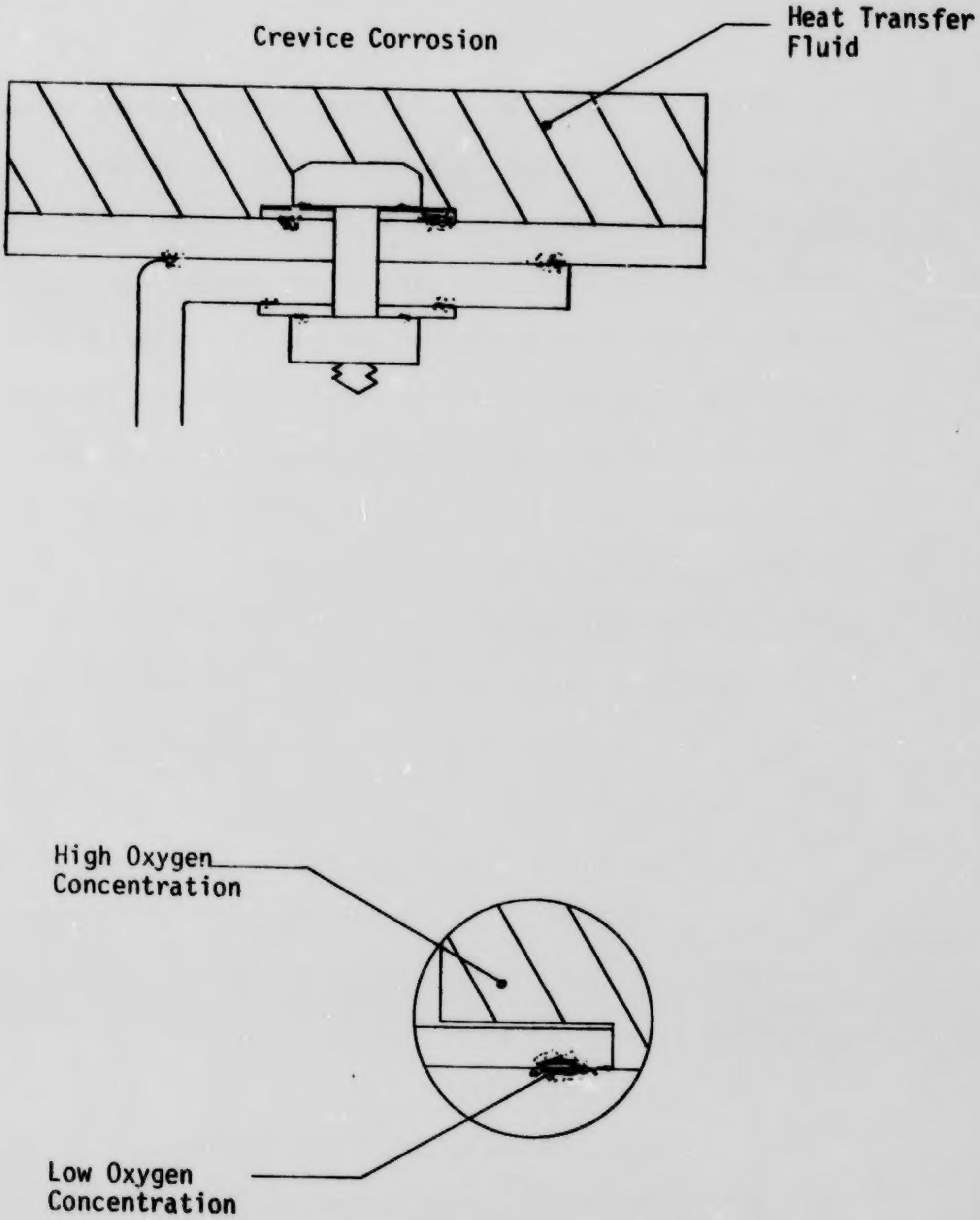
Crevice Corrosion
Heat Transfer Fluid
High Oxygen Concentration
Low Oxygen Concentration

FIGURE 7.3.2

Methods of Prevention - The most common control method is to use metals that are located close together in the galvanic series. Dissimilar metals (such as steel and cast iron) can be used in direct contact if they are located within the same set of brackets in Figure 7.3.1. Another control method is to avoid combinations of dissimilar metals where the area of the less noble metal is small compared to the more noble metal. It is also possible, of course, to electrically insulate dissimilar metals. A protective coating may be applied to the more noble end or chemical inhibitors may be added to the electrolyte.

2. Uniform Attack

This is a general corrosion which proceeds over the entire exposed surface. An example of this would be the simple oxidation of a metal which occurs in air.

To prevent this type of corrosion, durable materials should be selected, suitable inhibitors should be added to any liquids which are in contact with susceptible surfaces and protective coatings should be applied to exposed surfaces.

3. Crevice Corrosion

This is caused by the establishment of local corrosion cells. For example, the difference in oxygen concentration across the washer shown in Figure 7.3.2 can make the surface in contact with the low oxygen solution anodic and the other surface cathodic.

To prevent this corrosion, avoid rough spots where moisture, dirt and other contaminants can collect. Rough spots or rapid transitions in pipes should be avoided. The system should be designed to be completely drainable, thus avoiding pockets of stagnant fluid. Oxygen should be excluded from the heat transfer fluid - for example, through the use of air separators. Non-absorbent gaskets should also be used.

4. Pitting Corrosion

This is an extremely destructive form of localized corrosion which results in deep holes in the metal. Something will initiate a small hole in the material, and then corrosion will proceed rapidly due to the formation of a local concentration cell.

The use of chemical inhibitors (which form films on the metal) or complexing agents (which tie up metal ions in solution) may be used to prevent this type of corrosion. A getter column upstream from the metal which is to be protected can be used to act as a sacrificial anode. Another method is to use deionized water to remove chloride ions which can initiate the pitting. A good strainer should also be used.

5. Erosion Corrosion

This corrosion is a combination of chemical attack and destructive wear. It occurs where the fluid is travelling at a high velocity.

It can be prevented by modifying the shape and geometry of the system such that turbulence is reduced. The fluid velocity should also be limited. The National Research Council of Canada recommends a maximum velocity of 1.5 m/s for water or water solutions in copper tubing and 3.0 m/s for steel tubing. A good strainer should also be included to remove abrasives which can wear down the metal surface.

NOMENCLATURE

A	area, auxiliary, altitude
A_c	collector area
A_a	aperture area
A_r	receiver area
C	cost, concentration ratio, capacitance rate
C_p	specific heat
C_1, C_2	Planck's first and second radiation constants
C_b	bond conductance
D	diameter (defined locally), downpayment fraction
E	energy, equation of time
e	emissive power, base of natural logarithm
F	fin efficiency factor, control function (defined locally)
F'	collector efficiency factor
F"	collector flow factor
F_R	collector heat removal factor
F'_R	collector-heat exchanger factor
	annual fraction by solar energy
f_s	monthly fraction by solar energy, focal length, fraction of days with various K_T
g	gravitational constant
G_{sc}	solar constant, W/m
G	irradiance, W/m
H	daily irradiation, J/m
h	heat transfer coefficient, Planck's constant
I	hourly irradiation, J/m ; can also be considered as an hourly average rate
i	interest rate
K	extinction coefficient

$K_{\tau\alpha}$	incidence angle modifier
k	thermal conductivity, Boltzmann's constant
$\bar{K}_T$	monthly average daily clearness index
L	longitude, length, distance, loss, load
l	length
$\dot{m}$	flow rate
n	day of the year, index of refraction, hours of bright sunshine in a day
N	number of covers, day length, term of mortgage or economic analysis
P_{s_1}	dimensionless collector loss ratio
P_s	modified dimensionless loss ratio
P_L	dimensionless absorbed energy ratio
PWF	present worth factor
Q	energy per unit time, energy
q	energy per unit time per unit area or length
r	radius, ratio
R	ratio of total radiation on tilted plane to that in plane of measurement (usually horizontal), heat transfer resistance
$\bar{R}$	monthly average R
R_b	ratio of beam radiation on tilted plane to that on plane of measurement (usually horizontal)
S	absorbed solar energy per unit area
T	temperature
U	heat transfer coefficient
V	specific volume
W	distance between tubes
x	inflation rate
y	effective interest rate (i-x)

Greek

α	absorptance, thermal diffusivity
γ	surface azimuth angle, bond thickness, intercept factor
γ_s	solar azimuth angle
δ	declination, thickness (defined locally), dispersion
ε	emittance, effectiveness
η	efficiency (defined locally)
θ	angle (defined locally), angle between surface normal and incident radiation
θ_c	acceptance half-angle of CPC
λ	wavelength
μ	absolute viscosity, cosine of polar angle
ν	kinematic viscosity, frequency
ρ	reflectance, ground reflectance, density
σ	Stefan-Boltzmann constant
τ	transmittance, time
ϕ	latitude, angle (defined locally), utilizability
ω	hour angle, solid angle

Subscripts

a	air, ambient, absorbed, aperture, annual
b	blackbody, beam, back, bond, bed
c	collector, critical, cover, corrected
d	diffuse, day
e	effective, equivalent, edge
f	fin, fluid, fuel
i	incident, inlet
L	loss
n	normal, noon

p	plate
r	radiation, reflected, receiver
T	tilted, on a tilted plane
u	useful
z	zenith
λ	wavelength